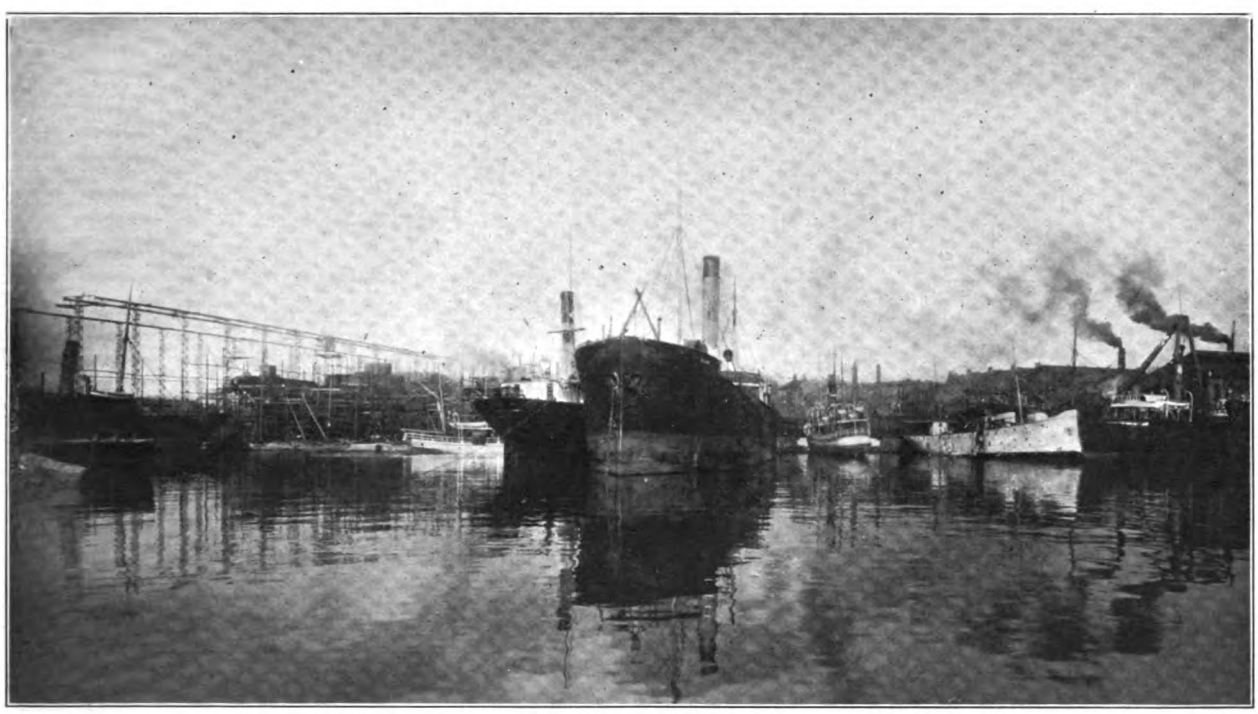
International Marine Engineering

JULY, 1909.

THE SHIPBUILDING AND ENGINEERING COMPANY OF AKERS MEKANISKE VERKSTED.

BY A. GUNDERSEN.

This company was established in 1842, and originally employed only a few men in a little workshop on the bank of the Akers River, near the city of Kristiania, Norway, from which river the company derived its name. In 1854 it was decided by its three owners (Messrs. Steenstrup, Schiött & Dybwad) to form a stock company and to move the works to the water front of the city of Kristiania, and to establish a shipbuilding yard in connection with the new and improved works, and also to change the motive power from water to steam. In the latter part of the same year they undertook to build their first steamer, which also was the first steam vessel built in Norway entirely of domestic make, as all the machinery parts were cast, forged and machined at the new works. From this unpretentious beginning grew gradually the present large company.



GENERAL VIEW OF AKERS MEKANISKE VERKSTED FROM THE WATER.

Up to 1867 the establishment had built and delivered forty steamers and ninety-nine engines. The subject of drydocks had by this time received little attention, and as the docking of ships is an indispensable part of their up-keep, it was decided in 1871 to build a graving drydock of dimensions to correspond with the demands of the port. The adjoining property was bought, work commenced, and the first vessel docked in 1874. Since then thousands of vessels have been docked and repaired in it, and the expectation of a profitable addition to the works has been fulfilled.

Up to the 1st of May, 1892, in which year the establishment celebrated its fiftieth year of existence, having built and delivered 137 steamers and 347 steam engines with boilers and auxiliaries, the attention of the firm had, on account of room and facilities, been directed principally to the building and repairing of smaller passenger and cargo steamers, and also to the building of steam whalers, of which the firm made a specialty, building a large number for Japan, England and America. In 1904 the desirability of improved and enlarged facilities was again considered by the board of directors, in order to meet the demands of the country for the building of larger vessels. Hitherto it could not be said that Norway's shipbuilding industry had kept up with the demands for larger vessels, and the many scattered shipyards in the country did not attempt to compete with other countries in the building of larger vessels. The result of the discussion by the board of directors in reference to enlarged facilities was an application to the government for adjoining land on which to extend the shipbuilding yard, and by a special act of the "Storting" the application was granted, and the work of enlarging and improving the plant commenced in 1906. It will thus be seen that the Akers Mekaniske Verksted has made successive extensions to meet a steadily increasing business, until now the plant occupies an area of about 8 acres with a waterfront of 1,000 feet.

In the last two years the motive power has been changed from steam to electricity, and Akers Mekaniske Verksted is

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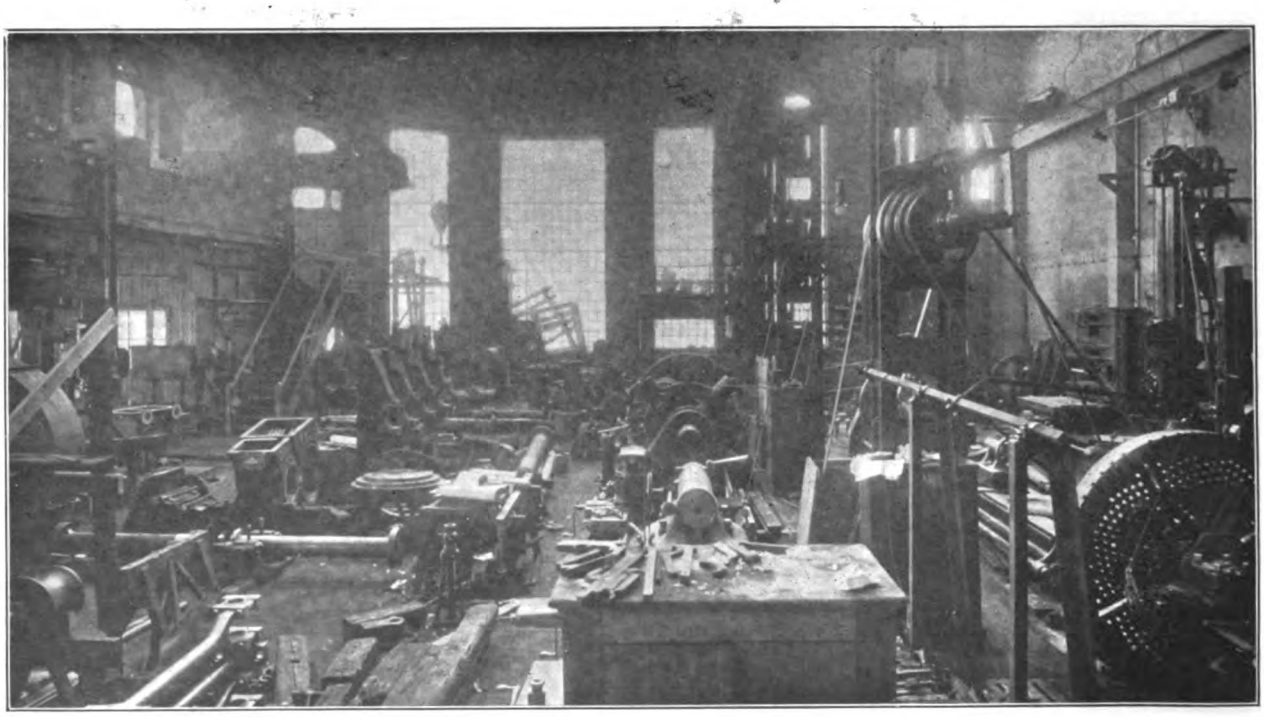
VIEW IN THE SHIPYARD.

perhaps at present one of the few shipyards in the world in which the motors are driven by electric current produced at a waterfall. The Kykkelsrud power station belongs to a private company, and is located on the bank of the River Glommen, about 50 miles from the city of Kristiania. From the power station the current, at 50,000 volts, is transmitted to the city, and transformed to a lower voltage at different stations and used for power and light throughout the city. The voltage, as the current enters the establishment, is 5,000, and in a special fireproof room, in which the transformers are located, the current is further reduced to 230 volts for power and to 110 volts for lighting. All the large tools in the different departments are driven by separate motors, and the smaller tools in groups from electrically-driven shafting.

From the plan of the works it will be seen that the shipbuilding yard now has five building berths, ranging in length from 490 feet to 150 feet, with plenty of water in front of them, and a bay large enough so that the vessels at launching do not have to be checked, a tugboat being sufficient to pick them up and bring them back to the works. The building berths have all necessary derrick cranes and winches for handling the materials, and the large berth is fitted with a skeleton steel frame structure on both sides of the berth 410 feet long, with I-beam runways on top, about 65 feet from the



VIEW IN THE MACHINE SHOP.

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ground, on which a trolley hoist is operated by electricity, with a lifting capacity of 6,000 pounds, a traveling speed of 200 feet per minute and a hoisting speed of 100 feet per minute.

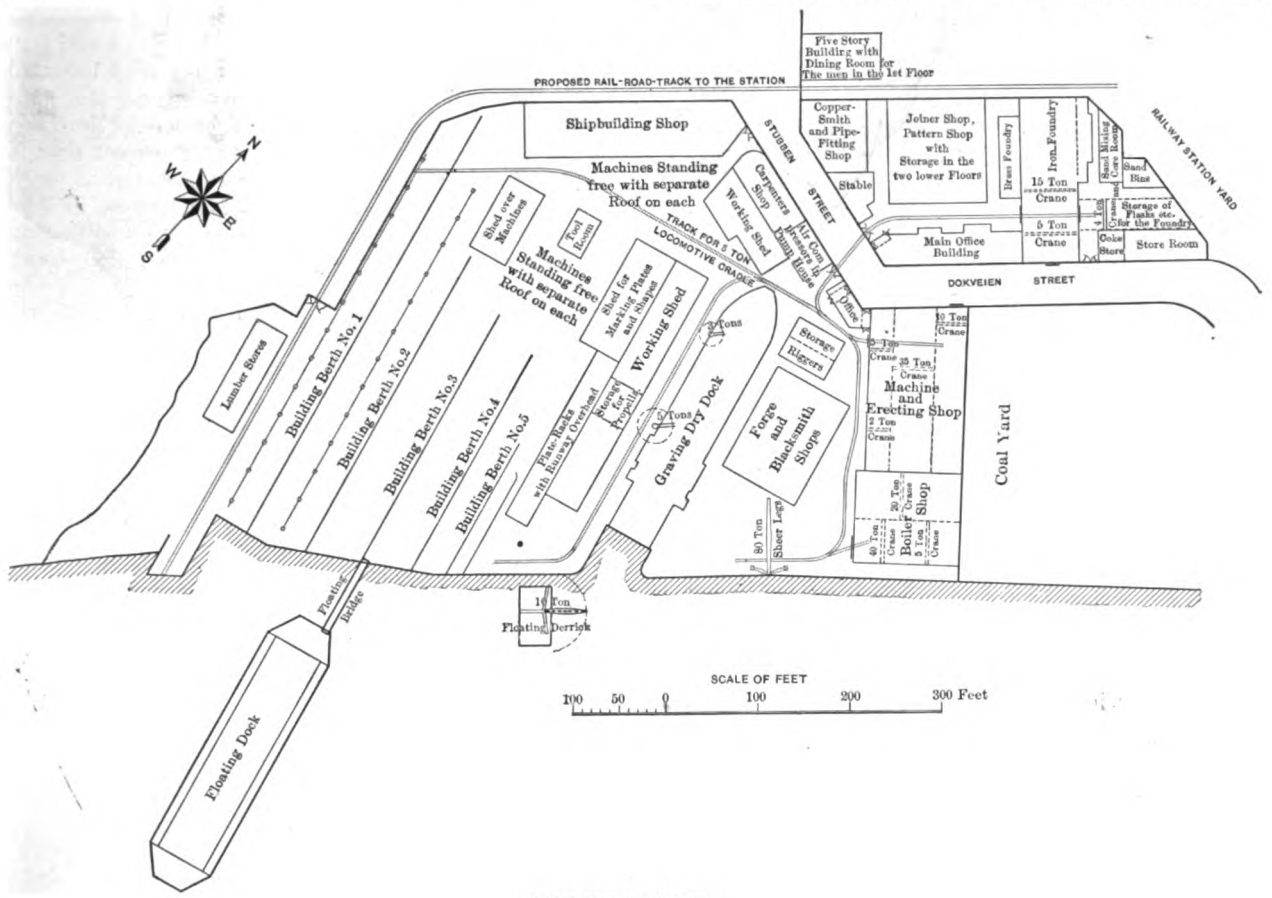
At the end of the building berths are located all the ship building tools, such as punches, shears, drills, countersinkers, beam benders, plate bending rolls, etc. The plate and angle furnaces, with bending slabs, garboard bender, scrieve-board, etc., are located in the building marked shipbuilding shop, with the mold loft in the second story of the same building.

For transportation of materials and articles between the different departments a wide gear system of rails is laid throughout the plant, on which a 5-ton steam locomotive crane is operated.

Lifting capacity about 2,200 tons dead weight.

The dock is provided with five centrifugal pumps, driven by electric motors, and four winches for handling and hoisting purposes, and it is in every respect of modern design and construction. The bottom of the dock has five independent pontoons bolted to the sides, and, as will be seen, the width is large enough to allow increasing the length whenever the demands of the port require it.

The Chicago Pneumatic Tool Company's system of air compressors is employed, and located as indicated on the plan, with pipes leading to all the different shops. One of the compressors is driven by steam and the other by electricity. There are also in use about 110 tools of the Chicago Pneumatic Tool



PLAN OF THE WORKS.

The graving drydock, previously mentioned, is of the following dimensions:

	Feet.
Length on top and bottom	281
Width on top	55
Width on bottom	38
Draft of water over sill	13

The pumping machinery consists of two 15-inch steam centrifugal pumps and two 6-inch pumps for drainage, driven by electricity, all of which are located at the end of the dock.

The floating drydock anchored in front of the building berths is new this year, having been built by William Hamilton & Company, Port Glasgow, Scotland, and is of the following dimensions:

	Feet.	Inches.
Length over outriggers	300	..
Length over pontoons	240	4½
Width outside	69	1½
Width inside at top of pontoons	55	..
Draft of water over keel-blocks 4 feet high at normal water-mark	15	..

Company's make, compressed air being extensively used in the shipbuilding yard for drilling, reaming, calking and riveting.

The carpenter shop is located next to the shipyard, in the same building as the dock pumps and air compressors, and is provided with the usual machine tools for rapid execution of the work.

The machine and erecting shops are partially steel and partially brick, 165 feet long and 100 feet wide, with three bays. The roof construction provides an ample supply of light in addition to the side and end windows. The three bays are divided from each other by two rows of steel columns, giving easy communication from one to the other. The west bay is fitted with a gallery throughout the length, on which a large number of smaller tools are located, and there are traveling cranes both under and over the gallery. The center bay, where the heavy tools are located, is provided with a 35-ton electric traveling crane with a height of 30 feet above the floor; this center bay is also used as an erecting shop, the entire ground floor being paved with wooden blocks on end. In the east bay are located all the medium-size tools. This

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is also provided with an electric traveling crane of 10 tons, at the same height from the floor as that in the center bay, for transportation of weights from one machine to the other. The machine shop building is new and modern in every respect, containing tools of the latest design and construction, built in Norway, the United States, Great Britain and Germany.

The boiler department adjoins the machine shop, and will be, when the new building is erected, a continuation of the machine-shop building and similar in construction, but without a gallery. The present boiler shop is provided with a 40 and a 20-ton electric traveling crane, running on beams at right angles to the cranes in the machine shop. Among the tools in this shop may be mentioned a large hydraulic riveting machine and a vertical plate bending machine, capable of bending plates 12 feet 6 inches in width and up to a thickness of 1¾ inches.

On the quay alongside the boiler shop is a set of 80-ton sheer legs, and a floating derrick with a capacity of 10 tons for handling and transporting the lighter weights between vessels and the quay.

The new forge and blacksmith shops will be located between the machine shop and the graving drydock in a single building, 115 feet long and 90 feet wide, and will be equipped with three steam hammers served by three jib cranes. The waste gases from the forge furnaces will be utilized to heat the feed-water for the boilers, and the exhaust steam from the hammers will be utilized for heating the machine and boiler shops during the cold season of the year. The blacksmith fires will be located along the side walls, and the air blast for the furnaces and fires will be provided by a fan driven by an electric motor.

The main offices are located on the opposite side of the street from the shipyard and machine shop, in a two-story and basement brick building, and incorporated with them are the drawing offices, where the ships and machinery are first designed, and where afterwards models and working drawings are prepared.

The foundries, joiner and pattern shops, pipe fitting and coppersmith shop, stables, storerooms, etc., are located on the office side of the street. The iron foundry is equipped with two Krigars cupolas, located on the east side of the building, and the blast is provided by a Root blower, driven by a 35-horsepower electric motor. The building is of steel frame work with brick filling, and is 172 feet long and 66 feet wide, with jib cranes at the sides and two electric traveling cranes operated on runways on top of the columns 23 feet from the ground. The brass foundry is located on the west side of the iron foundry, and on the east side of the same the sand bins, with mixing and core-making machines, are located, all driven by an electric motor as well as an open place for storing of pig and scrap iron and articles for use in the foundry.

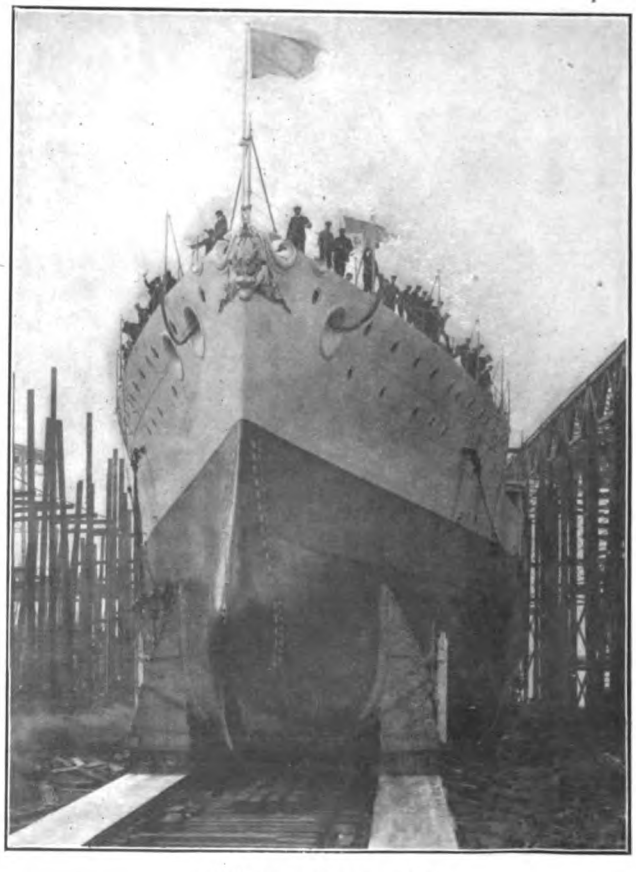
In the large three-story and attic fireproof building back of the office building the two lower floors are used for storage of materials and manufactured articles, and on the third floor are the joiner and pattern shops, equipped with modern woodworking machinery. In the attic is a well-arranged storage for patterns, and in the southwestern corner of this building is a large electric freight elevator for transporting the heavier articles up and down. The pipe fitting and coppersmith shops front on the street as well as the stable building, with living quarters for the stableman and janitor in the second story.

From the foregoing it will be understood that the works are to a great extent self-contained, as not only are ships built here but also the engines and boilers for propelling them, and all the various items of equipment and fittings required in the construction, including joinery, cabinet work and upholstery.

Up to the present time 286 hulls have been built and 572 marine engines with boilers, etc., including those under construction, of which the largest vessels have been of about 2,000 tons dead-weight capacity and the largest engines of about 2,000 indicated horsepower.

The art of shipbuilding in Norway has advanced of late years, and the products are comparable with those of other nations, but the cost and time of production of large-size vessels are perhaps greater than in other countries on account of the lack of materials, facilities to handle the heavier parts and the increased demand for skilled labor. With regard to labor economy, most of the different methods of inducing workmen to exert their full power have been tried, and the piece-work system is now extensively adopted, having proved very successful; but the premium system has not apparently been very widely used. The universal system of marking plates and shapes instead of making templets has also found its way to some of the Norwegian shipyards, and it must be stated, in justice to the men, that some of them are very good at it.

It is a well-known fact that the shipowners in Norway who have wanted larger ships have up to the present time, with a few exceptions, placed their orders elsewhere, as the Norwegian shipbuilding firms have not before seen their way clear to increase the productive capacity of their establishments to meet the demands for building larger vessels. This has been a great detriment to the Norwegian shipbuilding industry in the past, and the government has not assisted the home industry by giving a shipbuilding subsidy to encourage the undertaking in order to keep the business at home; but in spite of this, a few ship and engine-building firms in Norway, besides the one just described, have recently made improvements to meet the demands, and it is to be hoped that the nation will stand by them, and grant privileges for a few years, to assist in the undertaking of building larger vessels at home.



LAUNCH OF THE SAO PAULO.

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Launch of the Sao Paulo.

The Brazilian battleship *Sao Paulo* was launched on April 19 from the yards of Messrs. Vickers Sons & Maxim, at Barrow-in-Furness. The *Sao Palo* is one of three first-class battleships now building in England for Brazil. She is 500 feet long between perpendiculars, with a beam of 83 feet and a displacement on a draft of 25 feet of about 19,500 tons. The designed indicated horsepower is 23,500, and the speed 21 knots. Her armament consists of twelve 12-inch guns mounted in pairs in six turrets, four of which are on the center line of the ship. The secondary armament includes twenty-two 4.7-inch rapid-fire guns and eight 3-pounders. The heaviest armor is 9 inches thick, extending from well below the water-line to the upper deck. The ship is particularly well protected in the extent and distribution of her armor. Propulsion is by means of two four-cylinder, triple-expansion engines operating at 140 revolutions per minute. Steam is supplied by eighteen Babcock & Wilcox watertube boilers at a pressure of 250 pounds.

SUPERHEATED STEAM IN MARINE WORK.—I.

BY F. J. ROWAN.

INTRODUCTION OF SUPERHEATED STEAM IN MARINE WORK.

Considerable interest attaches to the fact that a large proportion of the experience gained from the use of superheated steam, at the time of its introduction some fifty years ago, was obtained in marine practice. Due to the limited development of boiler construction at that time, the steam pressures used in marine engines were low, and, consequently, the temperatures and range of expansion were low, so that only a moderate amount of superheat could be conveniently applied. This, of course, limited the advantage to be derived from superheating, and left room for a very ready and obvious means of obtaining the same results without superheating, viz.: by increasing the pressure and therefore the temperature of the steam in the boilers. Other considerations, such as the introduction of compound engines, troubles with superheater tubes and troubles with rubbing surfaces of cylinders, piston rods, valves and packings, induced engineers generally to abandon low-pressure superheated steam for high-pressure saturated steam. Nevertheless, engineers were familiarized with the general features of the use of superheated steam, and specially with the fact of the advantage to be derived from hotter, and therefore drier, steam.

There are three ways in which superheaters may be applied, examples of all of them being found in marine work:

1. They can be fitted in the flue space or up-take of boilers, so as to absorb heat from the hot gases after the gases have given up the larger portion of their heat to the boiler surfaces.

2. They may be placed in the boiler, forming really a part of the boiler construction and meeting with the hot gases at an earlier point than is possible with some of those which are placed in the up-takes.

3. They may be an entirely distinct apparatus, constructed with a separate furnace for independent firing.

In early marine practice only the first two methods were employed, the majority of superheaters having been arranged in the up-take or at the base of the funnel. One of the earliest was introduced by Messrs. John Penn & Son, in the P. & O. Company's steamer *Valetta*, and is shown in Fig. 1. The engines were of 260 nominal horsepower, and the boilers were of the "box" type, of Messrs. Lamb & Summers' design, the superheaters being placed in the up-take outside the ends of the vertical flues, which in Lamb & Summers' arrangement took the place of the usual horizontal flue tubes. The superheater was formed of horizontal wrought iron tubes, 2 inches inside diameter and 6 feet 3 inches long, arranged in two bundles or groups, each group consisting of forty-four tubes. These were placed in vertical rows, with clear spaces between the rows horizontally for access to the boiler flues for cleaning or brushing. The tubes were fixed at the ends into three flat chambers, made of wrought iron, welded up at the corners and each closed with a single flange joint. Steam from the boiler was admitted to the center chamber through a stop-valve, and was taken off from the end chambers by other stop valves communicating with the steam pipes to the engine. The total area of superheating surface, including the wrought iron chambers, was 374 square feet in each of the two boilers. The pressure of steam then used was 20 pounds

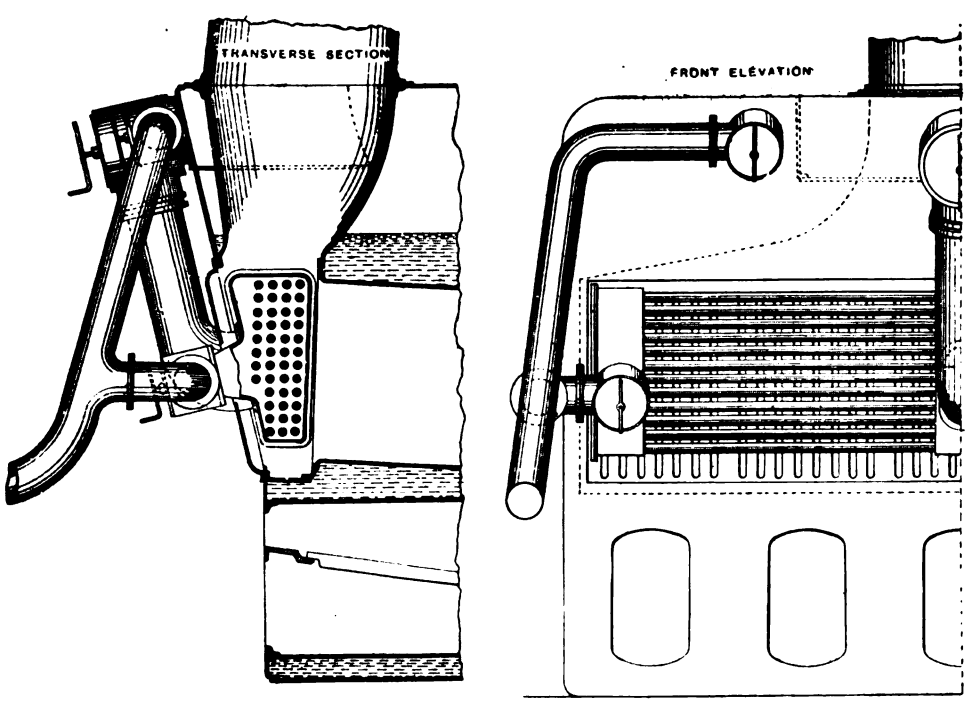

FIG. 1.—EARLY TYPE OF SUPERHEATER, INSTALLED ON P. & O. STEAMER VALETTA.

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per square inch, and the steam was superheated 100 degrees, or from 260 degrees F. up to 360 or 370 degrees F.

A different construction of the early up-take superheater is shown in Fig. 2, which was that of Patridge, fitted in H. M. S. *Dee*, in the R. M. S. *Tyne*, and in the Cunard Company's

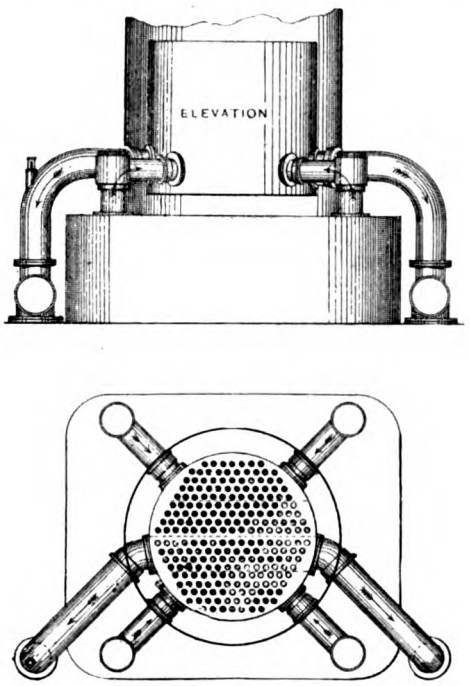

FIG. 2.—PATRIDGE UP-TAKE SUPERHEATER.

steamer *Persia*. A modification of it was introduced into the steamship *Great Eastern*. This superheater consisted of a cylinder filled with vertical tubes, and placed vertically over the up-take, resting on the steam chest at the base of the funnel. The hot gases passed up through the tubes and through an annular space surrounding the cylinder between it and the chimney; and the steam was passed across the cylin-

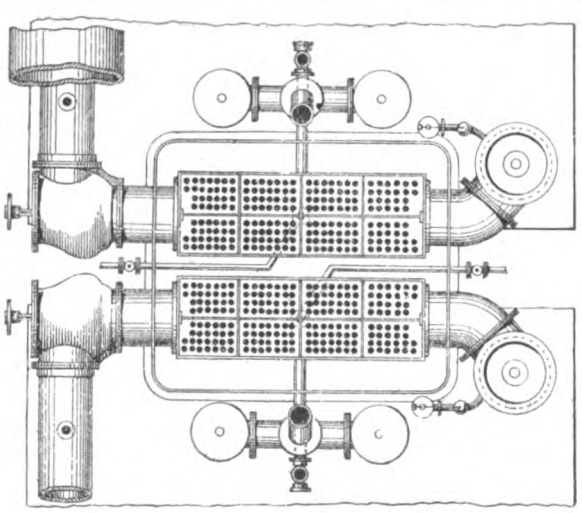

FIG. 3.—SUPERHEATER USED ON THE GREAT EASTERN.

der and over a vertical baffle plate in the center, steam pipes being arranged on each side of the cylinder at its base. In the case of the *Great Eastern* the superheater, constructed by Boulton & Watt, had an oblong form, and the chambers containing the vertical tubes were placed in a casing of similar form, which constituted the base of the chimney. This is shown in plan in Fig. 3. The same firm introduced a more simple form of superheater in the Holyhead steam packets. In these the lower part of the funnel was surrounded by a steam casing, which was divided radially by six partitions, the steam being caused to alternately ascend and descend in these until it passed over all the surface which was exposed to the heat from the chimney gases.

Fig. 4 shows another form of up-take superheater which was fitted in the steamer *Oleg* by Messrs. R. Napier & Sons. It consisted of horizontal steam tubes, 2 inches outside diam-

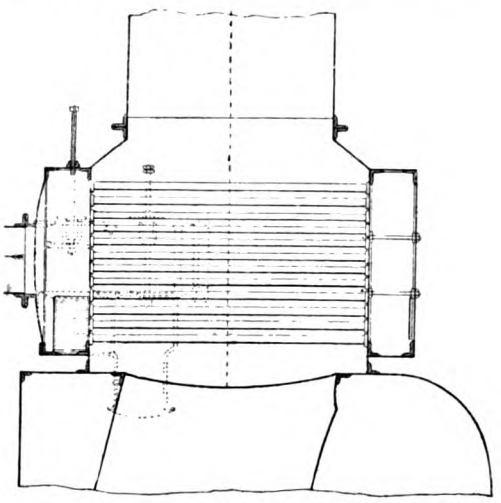

FIG. 4.—AN EARLY FORM OF UP-TAKE SUPERHEATER.

eter, 5 feet 6 inches long, fastened in flat, stayed boxes or headers, and placed in an oblong casing forming the root of the funnel.

Beardmore's superheater, Fig. 5, is an example of the construction as formerly arranged, in which the superheater forms a part of the boiler. In that instance no special stop valves were employed.

An entirely different arrangement, by which a much higher degree of superheat was obtained, is shown in Figs. 6 and 7. This was Parson & Pilgrim's superheater, which was placed in

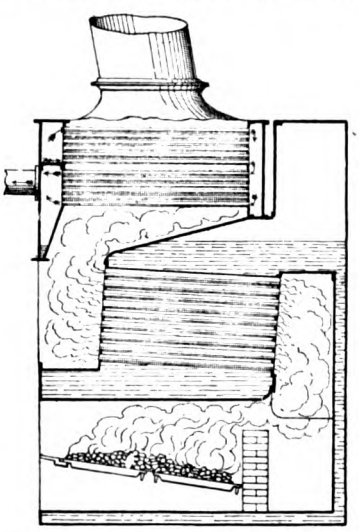

FIG. 5.—OLD-STYLE BEARDMORE SUPERHEATER.

the furnaces of marine boilers. A steam pipe, common to two furnaces, was led from the steam space to a position between the furnace doors, where it branched into two horizontal pipes, one of which entered each furnace below the fire-bars and passed nearly to the back of the grate. Two saddle-shaped pipes then rose from the horizontal pipe into the combustion space, and the steam passed through them to a return

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horizontal pipe on the opposite side of the ash-pit. The arched pipes frequently became red-hot, so that steam of 20 pounds per square inch pressure, or 264 degrees F. temperature, was found to have attained a temperature of from 484 degrees to 540 degrees F. After trial in a stationary boiler at Woolwich Arsenal, this superheater was applied to the boilers of vessels in Waterman's Steam Packet Company on the Thames and in H. M. steam tug *Bustler*.

Some other plans were proposed and tried in marine work, but accounts of their action have not been preserved. One deserving of notice, however, is that of Messrs. Wethered, on account of their endeavor to control the steam temperature by mixing saturated with superheated steam. Their superheater is said to have consisted of a coil of pipes, having about 3 square feet of surface per nominal horsepower, placed either in the combustion space or in the heating flues of the boiler; but in practice modifications of that arrangement were adopted, so that the superheaters fitted in the vessels of the P. & O. Company and in H. M. S. *Dee* were included in the number of those using Wethered's system,

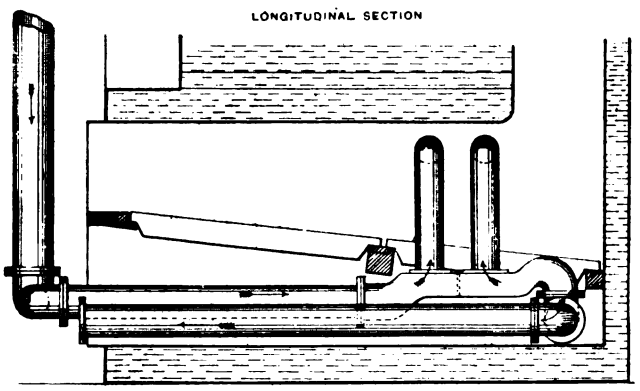

FIG. 6.

although the latter was also claimed as having been fitted with Patridge's superheater. Sir John Durston has, however, recorded that superheaters were brought to the notice of the Admiralty by Mr. Wethered, and trials were made in H. M. S. *Black Eagle* in 1856, which showed considerable economy in fuel in their favor. These and further trials resulted in their coming into general use about the same time as surface condensers; but a further step was made in 1860-63 by the introduction of compound engines with surface condensers and superheaters in H. M. S. *Constance*, a frigate of 500 nominal horsepower, which was fitted and tried in 1863. M. Felix Godard (Trans. Inst. N. A., 1908) states that the French navy tried superheated steam in one of their earliest protected cruisers.

Vessels belonging to the Russian Steam Navigation Company were included among those fitted with superheaters by John Penn & Son, along with others belonging to the P. & O. Company, and the saving claimed in several instances ranged from 23 to 34 percent.

In America trials were made in 1854, under the direction of Mr. B. F. Isherwood, in the steamer *Joseph Johnson*, with a mixture of superheated and saturated steam, and the results reported were extremely favorable. At a later date—1862-1864—the United States government instituted more thorough experiments, under Isherwood's direction, in the Bay line of steamers in Baltimore, principally in the *Eutaw*, and although an average gain in power was obtained of 18 to 20 percent it appears that the superheater pipes must have been seriously overheated at times, and the deterioration of these coils caused the trials to result in an unfavorable opinion of the practicability of superheating.

THEORETICAL CONSIDERATIONS.

The principal features of superheated steam may be summarized as follows: (1) Superheated steam is independent of pressure (that is to say, saturated steam of any pressure may be superheated) and admits of variation in temperature, while the pressure remains constant; (2) the temperature of superheated steam may be reduced without condensation taking place; (3) superheated steam is greater in volume than saturated steam of the same weight, but 1 pound of superheated steam contains more heat than 1 pound of saturated steam; (4) superheated steam practically follows the laws of a perfect gas.

To illustrate the relation which heat bears to the production of superheated steam, any table of the properties of saturated steam will show that such steam at a total pressure of 100 pounds per square inch has a temperature of 327.9 degrees F., and that the total heat of evaporation of 1 pound at that pressure is about 1,181 B. T. U., representing the sensible and latent heat employed. To increase the temperature of that 1 pound of steam by superheating 200 degrees F. at constant

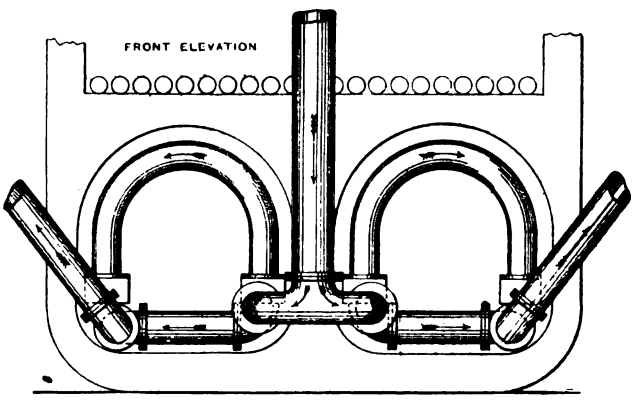

FIG. 7.

pressure, the addition of only $200 \times .520$ or 104 B. T. U. is required. The full value of this heat, as so applied, can be estimated only by its effect on the complete system, including both boiler and engine, but a rough comparison may be made by estimating the saving in coal at the boiler. The following example is said to correspond fairly with actual results: Assuming that a boiler evaporates 10,000 pounds of water per hour into steam of 165 pounds working pressure, from water at 64 degrees F., this requires 1,250 pounds of coal, taking 8 pounds of water per pound of coal as the rate of evaporation. This quantity of steam would supply an engine of, say, 450 indicated horsepower, using 22 pounds per indicated horsepower-hour; 70 percent of the saturated steam is taken as doing useful work and 30 percent as wasted by condensation, etc. If the steam were superheated 200 degrees F., cylinder condensation and other troubles would be practically avoided, and the volume of each pound of steam would be increased by 25 percent. Taking this figure to represent the gain, the same horsepower could be developed by $\frac{10{,}000}{1.25} = 8{,}000$ pounds steam when superheated. At the same rate of evaporation per pound of coal 1,000 pounds of coal per hour would be required, plus the quantity required to superheat the steam. This would be

$$\frac{200 \times .541 \times 8{,}000}{8 \times (1{,}196 - 32)} = 93 \text{ pounds per hour.}$$

Thus the saving in coal works out at 1,250 — 1,093 = 157 pounds, or 12.56 percent.

The advantage to be derived from superheating is in this

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illustration supposed to depend solely upon the increased volume of the steam, but that is not strictly correct. It is now agreed that the thermodynamic advantage is comparatively small, and that practically the whole benefit from superheating is due to the prevention of cylinder condensation and leakage.

Calculations founded upon the *pv* diagram are unsatisfactory, because of uncertainty as to the proper index of the adiabatic expansion curve; and the theoretical conditions represented on the Θ Φ diagram are almost impossible of realization. The true test of the advantage of superheating is that of a heat balance as applied to the entire system, including boiler, superheater and engine, or of thermal units per horsepower-minute or hour.

In all calculations connected with the use of superheated steam, or the capacity of superheaters, the specific heat of the steam is a most important factor. Regnault's figure of 0.48 was for years relied upon as being a correct mean value, but investigations carried out in Germany by Knoblauch and Jakob, and in America by Drs. Thomas, Heck and others, have shown that there is a considerable variation in the specific heat with altered conditions of pressure and temperature. It is found that, generally, the value of the specific heat decreases for any *pressure* as the temperature rises, and increases for any given *temperature* as the pressure rises.

In Fig. 7 a diagram is reproduced showing isobaric curves of the specific heat at constant pressure of superheated steam, derived from Knoblauch and Jakob's experiments, and the following table gives the mean specific heat for superheat from saturation temperature t to t_s degrees. The figures are given in French and British units:

MEAN SPECIFIC HEAT FOR SUPERHEATS FROM SATURATION TEMPERATURE t TO t_s DEGREES.

p. in kg. pr sq. cm.	1	2	4	6	8	10	12	14	16	18	20
p in lbs. per sq. inch	14	28	57	85	114	142	170	199	227	256	284
t in dgs. C.	99	120	143	158	169	179	187	194	200	206	211
t in dgs. F.	210	248	289	317	336	354	369	381	392	403	412
t_s Degrees Fah.	t_s Degrees Cent.											
212	100	.463	...									
312	150	0.462	0.478	0.515								
392	200	0.462	0.475	0.502	0.530	0.560	0.597	0.635	0.677	(0.751)		
482	250	0.463	0.474	0.495	0.511	0.532	0.552	0.570	0.588	0.609	0.635	0.664
572	300	0.464	0.475	0.492	0.505	0.517	0.530	0.541	0.550	0.561	0.572	0.585
662	350	0.468	0.477	0.492	0.503	0.512	0.522	0.529	0.536	0.543	0.550	0.557
752	400	0.473	0.481	0.494	0.504	0.512	0.520	0.526	0.531	0.537	0.542	0.547

Figures have been published by Thomas in America and by Mollier in Germany which do not materially differ from those of Knoblauch and Jakob. Mollier gives the following formula for the specific heat for any degree of superheat:

$$(C_p)^t_{t'} = \frac{H - H'}{t - t'}$$

where H = the total heat in the steam at point of superheat, H' = the total heat in the steam at point of saturation, and t and t' are the respective total temperatures.

From a recent research by Dr. Harvey N. Davis, of Cambridge, Mass., on the total heat of saturated steam, it appears that Regnault's figures and formula are not correct, due probably to the presence of unobserved moisture in his experiments. New tables of total heat, specific volume, etc., of steam are therefore to appear, but meantime engineering calculations will not be seriously affected by the use of the old values in existing steam tables.

Estimates of theoretical economy, due to the use of superheated steam, vary greatly according to the basis upon which economy is reckoned. There may be, and have been, for instance, estimates of the quantity of steam used by an engine showing a saving in *steam*, due to superheating of 14 percent and upwards, according to the kind of engine—reciprocating or turbine, high speed, etc.—employed. The saving in *heat* from the point of view of preventing initial condensation by superheat may be only some 3.5 percent; but the total saving in heat is a much larger question, and includes savings in generating steam, preventing condensation in pipes, cylinders and valves and diminishing leakage, all of which can be estimated only from the performance of the whole apparatus. The saving in coal cannot be directly estimated from the superheating, apart from the use of the steam, as has been remarked. In the case of steam turbines the use of superheated steam has been proved to effect a reduction of fluid friction, and this is an element of economy in their working.

The following two tables show percentage reductions in steam consumption, due to superheating, which have been obtained in turbines and in reciprocating engines in land

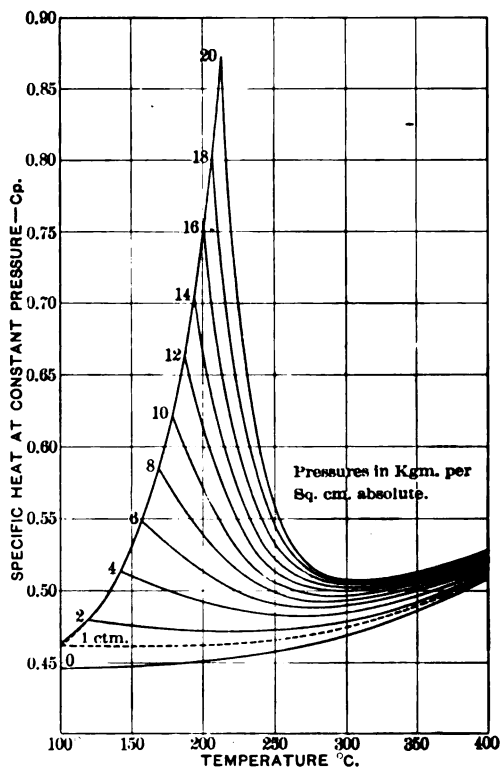

FIG. 7.

installations. These were carefully collected by Mr. R. M. Neilson, and given to the Cleveland Institute of Engineers. Of course, it is necessary to know the details of each test in order to be able properly to understand the relative value of the reduction of steam consumption in each case.

TABLE A.

REDUCTION IN THE STEAM CONSUMPTION OF TURBINES DUE TO THE USE OF SUPERHEATED STEAM.

Degrees Fahrenheit of superheat	13	50	60	66	70	84	100	140	150	200	260
Percentage reduction of steam consumption	6.1	8.0	5.4	2.1	7.5	7.7	14.0	12.6	19.7	23.0	24.5
Percentage reduction per degree Fahrenheit	0.47	0.16	0.09	0.18	0.11	0.09	0.14	0.09	0.13	0.115	0.09

TABLE B.

REDUCTION IN STEAM CONSUMPTION OF RECIPROCATING ENGINES.

Degrees Fahrenheit of superheat	31	40	50	100	150	216	225	225	440
Percentage reduction of steam consumption	7.86	8.65	12.00	20.55	13.00	36.4	33.7	33.1	30.9
Percentage reduction per degree Fahrenheit	0.25	0.22	0.24	0.20	0.09	0.17	0.15	0.15	0.07

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In marine practice, from the nature of the service—except in the case of warships or in specially observed trials—results are not collected with the same care as on land, and hence economies are more roughly estimated. The total expenditure of coal, or the radius of action, is usually the test applied. So far, superheated steam has been introduced only to a small extent with the use of turbines on board ship, but there is no reason why its advantages with turbines on land should not be realized in marine work also. There are some successful examples which will be noted later; but the larger number of steamships hitherto fitted with superheaters have reciprocating engines.

On its first introduction, about fifty years ago, it was proposed that superheated steam should be used in a mixture with a certain proportion of saturated steam, and it is interesting to find this proposal revived in a discussion at the Institution of Naval Architects last year. The advantages of this mixture were supposed to be twofold, viz.: to effect control of the temperature of the steam admitted to the engine, and to provide lubrication of the rubbing surfaces by the condensation of a small quantity of the steam. The plan of Messrs. Wethered, which depended upon this mixture, was, however, to a great extent deprived of its exclusive value by the discovery that when a temperature of not over 360 degrees F. was used no admixture of saturated steam was necessary. To-day the proposal to use this mixture seems to be with a view to control temperature at the superheater, but it has been pointed out that spraying water into the superheater tubes is more efficacious. The former difficulties of using steam of high superheat in engines and of lubrication have almost entirely disappeared.

INFLUENCE OF THE USE OF SUPERHEATED STEAM ON ENGINE DESIGN.

The main reason for modification in the design of engines to use superheated steam is that the effects of high temperature must be provided for. With a so-called "moderate" superheat, or about 150 degrees F. at the boiler, which yields about 100 degrees F. at the engine, practically no alteration would be required, but the superheat disappears by the end of the stroke of the high-pressure piston, and if, in a compound engine, it is desired to have superheated steam in the low-pressure cylinder, the high-pressure exhaust steam must be passed through a reheater. It is not difficult to understand that the use of superheated steam has shown that with it there is no advantage to be gained by multiplication of cylinders—a compound or triple-expansion engine providing all that is required for economical working. The real advantages of superheated steam are, however, it is maintained by many, to be obtained only from what is called high superheat, which varies from 200 degrees F. upwards. When that temperature, or even 150 degrees F. at the engine cylinder, is added to the temperature, due to saturated steam of high pressure, a degree of heat is reached which introduces special conditions of expansion, erosion, tenacity and ductility, condition of surface, etc., in the different metals ordinarily employed, and these affect pipes, cylinders, pistons, valves, etc., and also packings and lubricating material.

Generally, as a well-known authority has expressed it, the steam pipes should be of lap welded or drawn iron or steel, with ample provision for expansion, and where there is excess of boiler pressure they should be of smaller diameter than is usual when saturated steam is used. They should be carefully lagged and the flanges should be covered by removable boxes. The cylinders should be designed to allow the parts exposed to the entering superheated steam, the working steam and the exhaust steam to expand independently of each other. Care is required in the selection of the quality of metal for the cylinders; which should have uniformity of thickness, without projections, as far as possible, to favor equality of expansion throughout. The dimensions of the cylinders will be increased if their number is reduced, and the speed of working can be increased with superheated steam. The pistons also require care in making, and should be carefully guided and lubricated directly rather than by mixing oil with the steam. Corliss valves can be used with moderate superheat, but higher degrees require drop-piston valves, or double-beat equilibrium valves working vertically; and no metals should be employed which have a melting point within reach of the steam temperature. The packings of piston rods and valve spindles should be metallic, and lubricating oil should be of the mineral variety, such as is employed in gas engines, in which far higher temperatures are successfully dealt with. Scoring of cylinder surfaces, pistons and piston rods has been found to emanate from the decomposition of unsuitable lubricants, and even trouble with piston rings has been discovered to be due to the same cause. With proper attention to the form and distribution of metal in cylinders, valves, etc., the chances of any of the moving parts seizing are reduced to a minimum, and the friction of the engine will be kept low.

The reduction of leakage and condensation is due to the quality of the steam, and is greatly in favor of the smooth working of the engine; a more even distribution of steam during the stroke is given, and a reduced total weight of steam being used the work of the air pump and condenser is rendered lighter. The advantage reacts even upon the boiler, a reduction in the weight of steam required meaning either fewer boilers or easier firing.

The part of the apparatus which apparently needs the most care is the superheater itself, as it is exposed to the danger of overheating. The methods of construction now in use, and the care taken in arranging the position of the superheater relatively to the temperature of the hot gases reaching it, have, however, greatly discounted that risk, and instances are not wanting of superheater tubes lasting practically as long as the boiler. Marine practice is liable, from more frequent stopping and starting of the engines in some kinds of service, to put a more severe strain upon the superheater as regards fluctuations of temperature than is work on land; but this, as other conditions, can be successfully encountered.

The question of reheaters and jackets would require a large amount of space for its full discussion. Theoretically, it is better to superheat to a high temperature than to a lower one and to use reheaters to carry the necessary superheat to the later stages of the engine or turbine. But practical difficulties with the higher temperature turn the scale in favor of the less perfect method. In fact, in the Schmidt system of using steam of high superheat, the steam is passed through the tubes of a reheater standing in the path of the exhaust from the high-pressure cylinder before it enters the high-pressure cylinder, and thus the temperature of the steam is modified even in that cylinder. With both reciprocating compound engines and with multiple-stage turbines it is common, in land practice, to use reheaters between the cylinders or the stages of expansion. The turbine gains considerably in efficiency by the use of superheated steam, and the steam consumption in some land installations is stated to be reduced 1 percent for every 10 degrees to 12 degrees F. of superheat used, leading to a saving in fuel consumption of about half that amount. The high temperature, however, demands that the clearance over the tips of the blades must be made slightly greater to allow for extra distortion of the cylinder, although this may involve some slight loss by leakage. The discharge angle of the blades is also affected by the coefficient of friction being less with superheated than with saturated steam.

Finally, as to steam jackets, it seems to be settled that it is useless to jacket the high-pressure cylinder, using superheated steam, unless with the waste furnace gases, but that steam

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jackets may be used with advantage on the low-pressure cylinder in order to prevent condensation there, if the saturated condition is reached by the steam at any early period in the stroke.

(To be continued.)

SIMPLE METHOD OF PROPELLER DESIGN.

BY CHARLES S. LINCH.

In the design of a propeller wheel there are a number of assumptions to be made, and when data from other ships are at hand it is better to obtain such data from previous work for the purpose of comparison. In the delineation of the propeller wheel these things do not have to be considered, as it is the designer's place to give the draftsman the necessary information, but the knowledge of the necessary steps is of great importance, and hence, before proceeding to describe the methods, we will work out the computations and then proceed to explain the different methods of drawing the blade.

The wheel which we will take as an example is one designed by the writer for a freight and passenger ship. The requirements were as follows: The designed indicated horsepower was 1,989, and it was obtained from computations for the resistance, etc. The speed was 15.5 knots. The revolutions were to be between 86 and 88 per minute.

The developed area of the blade was obtained by first taking a standard elliptic blade with a mean width ratio of .2, and computing the area from this. For example, we know that the mean-width ratio is equal to the mean width divided by the diameter of wheel in feet; and, further, we know that the mean width is equal to the developed area of the blade, divided by the diameter of the wheel, minus the diameter of the hub, and this difference divided by two. From the draft of water aft and by making the proper allowance for immersion of the tip of blade, a diameter of wheel was assumed. The developed area was computed as equal to 19.1 square feet for each blade. The diameter of hub was made equal to 0.24 times the diameter of the wheel. As this wheel was of the built-up type, this gave a hub of ample dimensions.

The apparent slip, taken from the performance of a ship of similar dimensions, equals 10 percent. The true slip was 22 percent; therefore the wake factor was 16 percent. The propeller efficiency was taken at 65 percent and the engine efficiency at 87 percent; therefore the propulsive coefficient is .87 × .65, or 56 percent. The speed of the ship in feet per minute is equal to $\frac{15.5 \times 6{,}080}{60}$, or 1,570.15 feet per minute.

Now 1 — apparent slip = 1 — .1 = 0.9.
$.9PR$ = 1,570.15, therefore PR = 1,744.6.
The log of 1,744.6 = 3.2416959.
The log of $(1{,}744.6)^3$ = 9.7250877.
Corresponding number = 5309000000.
The log of 60 = 1.7781513.
The log of $(60)^3$ = 5.3344539.
Corresponding number = 216000.

Now, assuming a standard elliptic blade, and that $\frac{P}{D}$ = .76, we have for the characteristics of the blade:

A = 0.340.
B = 0.569.
C = 1.354.

With width ratio (mean) of 0.2 we have

A = 0.340 × 0.2 = .0680.
B = 0.569 × 0.2 = .1138.
C = 1.354 × 0.2 = 2.708.

Now the propeller power equals 1,989 × 0.56 = 1,113.84; therefore

$$I' = \frac{1}{550 \times (60)^3} P^3 R^3 D^2 N (1 - s) (a s A - f B).$$

Substituting and solving for D we have

$$D^2 = \frac{1{,}113.84 \times 550 \times (60)^3}{(1{,}744.6)^3 \times 4 \times .78 (2.74 \times .22 \times 0.068 - 0.016 \times 0.1138)} = 204.27$$

Therefore D = 14.28 feet, $\frac{D}{P}$ = 0.76; therefore P = 14.28 ÷ 0.76 = 18.8 feet. Since the mean-width ratio = $\frac{\text{mean width}}{\text{diameter}}$, we have 0.2 = $\frac{\text{mean width}}{14.28}$; therefore mean width = 2.856 feet, and 2.856 = $\frac{\text{developed area}}{\text{diameter wheel} - \text{diameter hub}}$, or

$$2.856 = \frac{2 \times \text{developed area}}{10.78};$$

therefore developed area = 15.39 square feet.

Now PR = 1,744.6.
P = 18.8 feet, say 19 feet, therefore,
R = 91.7.
We, therefore, have by computation:
Diameter wheel = 14.28 feet.
Pitch wheel = 18.8 feet.
Revolutions wheel = 91.7.
Developed area = 15.39 square feet.
Mean width ratio = 0.2.
Mean width = 2.856 feet.
Maximum width = 3.1 feet.
The wheel as built was 14.5 feet diameter, 19.0 feet pitch.
Developed area = 15.5 square feet.
Diameter hub = 3.5 feet.

We are now ready to draw the wheel. First draw the line $X X'$ as shaft center, and erect the perpendicular $Y Y'$, intersecting the line $X X'$ at O. Set off on $Y Y'$ the distance O-1 equal to the radius of the hub. Set off on $Y Y'$ O-7 equal to the radius of the wheel. Divide the distance O-7 into any convenient number of parts, and draw through these points horizontal lines parallel to $X X'$. Now place over this much of the drawing a piece of tracing paper, and sketch the outline of developed contour. If we are working from projected area, then this area is to be found and the shape corresponding. We will assume that we have the developed area as a basis. Now, with a standard blade (elliptic) we can divide the vertical distance O-7 on the tracing paper into either tenths or twentieths and compute the widths at these points. These dimensions will be parts of the maximum width, or if not computed we can follow the suggestions of Prof. Durand, namely, describe on one side of the vertical a rectangle of area near the required area, and sketch in free hand the contour.

After this is done, take planimeter readings, and after the area and shape are satisfactory we can measure the widths at the point of intersection of the horizontals 1, 2, 3, 4, etc., with the contour of the blade. We can now remove the tracing paper and transfer these measurements to the drawing.

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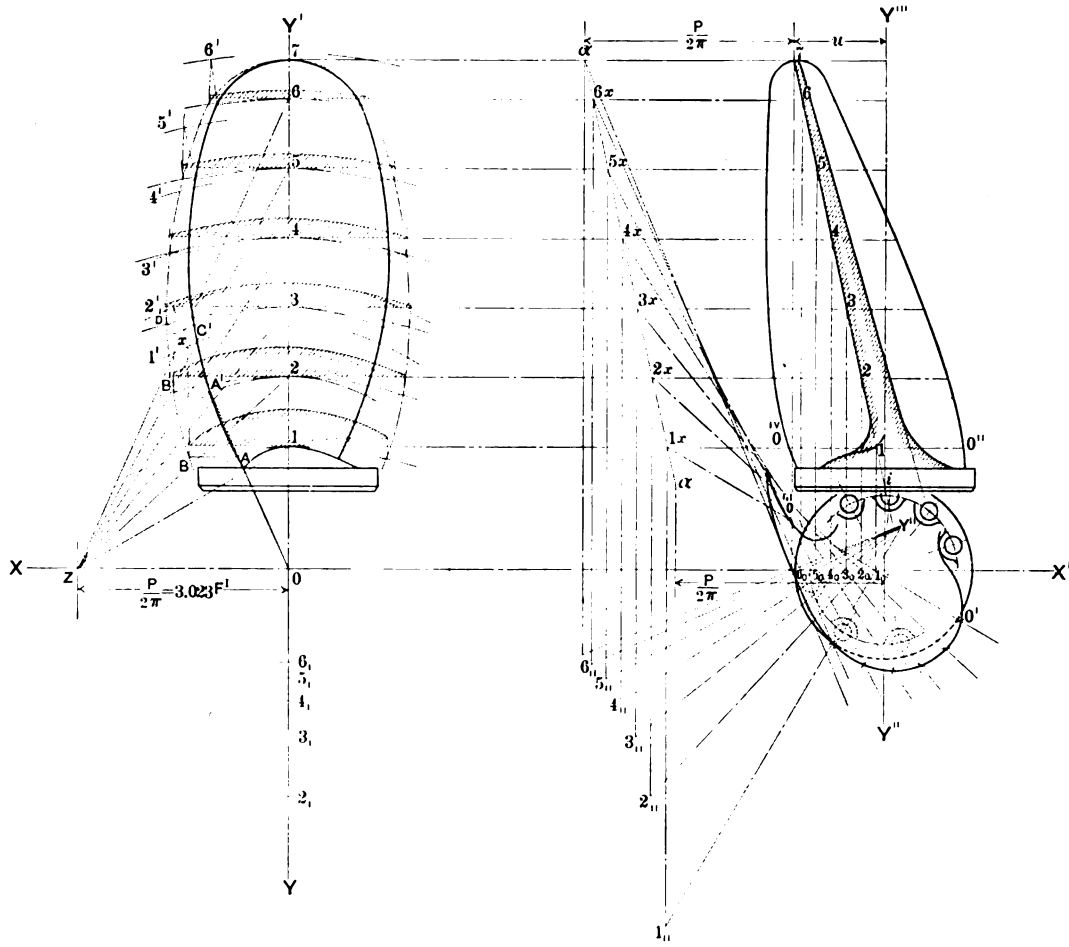
After these points are properly located, we can run a fair curve through them, and we have now the developed shape of the blade. This then is the 'thwartship view. At a convenient distance we erect another vertical line $Y''' Y'''$'. This is to be the fore-and-aft view of blade and hub. We now set off the rake u and through the points 1, 2, 3, 4, etc., we produce the horizontals to intersect the vertical $Y''' Y'''$'. After setting off the rake u, we draw a line from this point, intersecting the vertical $Y''' Y'''$' in i, in this case. This is the slope of the generating line. Some blades intersect at the point of intersection of the vertical with the horizontal. Now, from point 7 and point of intersection of $Y''' Y'''$' set off parallel to $X X'$ a distance equal to $P \div 2\pi$, draw through these points a straight line. This line is parallel to the generating line.

Now where the arcs $1B$, $2B'$, $3D'$, etc., intersect the developed contour of the blade, erect perpendiculars intersecting the arcs $1'$, $2'$, $3'$, etc., as at x. From this point of intersection draw a diagonal to the point o, and where this point intersects the circular arc as at A, this point is a point of the projected view. After marking off the points, as above described, a fair curve passed through them gives the boundary of the projected area. We can proceed in the same way for the other side, but, if the blade is symmetrical, about the center line $Y Y'$ we can step the distances off with dividers, or, better still, if the work is done very accurately, we can simply draw the horizontal line, such as $A B$, $A' B'$, etc. We now have the two views on the thwartship plane. Some prefer to draw in the hub first, but as the hub is of small consideration, it can be left until the different views are finished.



LAYOUT OF A PROPELLER BLADE.

From the points 1, 2, 3, 4, etc., let fall verticals intersecting the line $X X'$ in 1_o, 2_o, 3_o, etc. From the points 1_x, 2_x, 3_x, etc., let fall perpendiculars. From the points 1_x, 2_x, 3_x, etc., draw diagonal lines, passing through the points 1_o, 2_o, 3_o, etc.

We then draw the lines $1_o 1_{||}$, $2_o 2_{||}$. These lines are drawn at right angles, respectively, to 1_x, 1_o, 2_x, 2_o, etc.

Now the point of intersection of the verticals dropped from 1_x, 2_x, 3_x, etc., are the radii of the circular area $1B$, $2B'$, $3D'$, etc., on the athwartship plane. With the radii $1_{||}$ 1_x, we describe the arc, as above mentioned, passing through 1. With radii $2_{||}$ we describe the arc passing through 2. The center points are shown on $Y Y'$ at 2, 3, 4, etc. After these arcs are described, we proceed to describe circular arcs with radii o 1. o 2, o 3, etc., passing through the points 1, 2, 3, etc.

Set off from O the distance $P \div 2\pi$ and draw from the point Z straight lines to the points 1, 2, 3, etc. With $Z1$, $Z2$, $Z3$, etc., as radii and O as center, describe the small arcs $1'$, $2'$, $3'$, etc.

Taking the fore-and-aft view, we proceed as follows: With the dividers take the distance $1B$, and set this distance off on either side of the line 1_x 1_o, using the point 1_o as center, as shown with O' and O'''. With the length $2B'$ set off on either side of the line 2_x 2_o this distance and do the same for each point. Passing a fair curve through these points gives us the view when looking at the blade from the tip.

After these points are located, project up to the horizontal line, passing through $7i$. Thus the point O' is projected up to the line, passing through 1, and we obtain one point on the right of 1 as at O''. Projecting point O''' up to 1, we obtain point O^{IV}. Proceeding in this way for the different points, we obtain a series of points, and by passing a curve through these we get the contour of the blade in the fore-and-aft plane.

Having computed the thickness of the blade at the root and tip, set these distances off and draw a line through them, connecting the blade to the hub with fillets. It is now possible on

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the thwartship plane to show the shape of the blade. This needs no explanation, as it is self-evident.

Assume that we have given the projected dimensions. With a distance $P \div 2\pi$ set off from O, erect the pitch lines from Z to 1, 2, 3, 4, etc. With a radius equal to $O1$, $O2$, $O3$, etc., and with O as center, describe the circular arcs passing through 1, 2, 3, etc. With a radius $Z1$, $Z2$, $Z3$, etc., and O as a center, describe the circular arcs 1', 2', 3', etc. Where the circular arcs pass through the point on the projected area, draw a diagonal line to O, intersecting the arc 1'. From this point of intersection of the diagonal with the arc 1', drop a perpendicular. From the point of intersection, as at $A1$, on arc 1, draw a horizontal line $A\ B$, parallel with $X\ X'$, and where the horizontal line intersects the perpendicular, this point of intersection is a point of the developed contour of blade. Proceeding thus, we obtain a series of points, and by passing a fair curve through same, we obtain the developed contour. The fore-and-aft view is as described above. It is as well to show the outline of the palm of the blade, showing the location of the holes for studs, the elongations, etc.

While the method of elliptic arcs is the most workmanlike, yet the three methods give precisely the same results. The two methods described above are simpler and much easier to handle. The writer has used these methods for several years, and while, in a majority of cases, he has used the elliptic arcs, it was due to the pattern maker measuring and building his pattern as described. The proof of the correctness of the above construction is very tedious, and space will not permit of the mathematical demonstration here. If we lay down the elliptic arc we will see that the difference is so slight that it is only by mathematical proof that we are able to discern any difference. Before closing this article, the writer would like to try and impress, not only on draftsmen, but on designers in general, the necessity of very accurate data in reference to this most important part of marine engineering. In the use of this method full-sized drawings of the developed areas were made, and when the pattern was made these were tried over the blade. The shape was laid out by the three methods described and made to shrinkage rule. There was exact coincidence.

SPEED TRIALS OF THE DESTROYER COSSACK *

BY SIR PHILIP WATTS, K. C. B., F. R. S., LL. D.

In the discussion which followed the reading of Mr. J. L. Thornycroft's paper last year I stated that the Admiralty were taking steps to obtain by a series of comparative trials additional data concerning the effect of shallow water upon the speeds of ships.

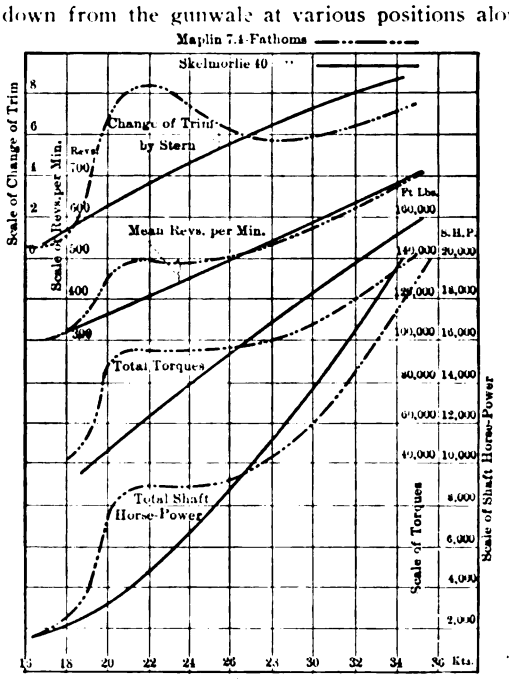
The trials have been carried out, so far as the exigencies of the service have permitted, and, although they are far from being as full as could be desired, they will probably be regarded as of sufficient interest to be placed on record in the Transactions of this institution.

A full series of measured-mile runs was made with the *Cossack* at speeds varying from 17 knots to 34.5 knots, first on the Maplin Sands, and afterwards at Skelmorlie (depth, 7.4 fathoms at the Maplins, and 40 fathoms at Skelmorlie). They were made under as fair conditions as can ordinarily be secured without undue expenditure of time and money in waiting for specially favorable weather. The conditions were practically the same as on the official trials of this type of ship, except that there were no restrictions as to the consumption of fuel. The Cossack was chosen for the purpose as she had recently been delivered, and her propeller shafts had been calibrated for torsion stresses.

Each shaft had Bevis-Gibson flashlight torsionmeters of the axial type, supplied and erected in place by Messrs. Cammell Laird. It was found possible to get a length of shaft of 20 feet between the two discs of the apparatus, so that the readings on the scale at the eye piece were fairly open and reliable; and as the torsionmeters, when once fitted, were not disturbed till the experiments were completed, any slight fluctuation of torque with revolution does not vitiate the comparison of the results obtained on the different trials.

Revolution counters, worked from each shaft, were fitted on the upper deck, and a 20-foot water level was erected in the middle line plane of the ship, to enable the change of trim to be accurately measured. The vessel was kept nearly at the same trim and displacement throughout the trials.

The speeds over the mile were carefully taken by independent observers. The wave profiles were obtained by measuring down from the gunwale at various positions along the



DATA FROM SPEED TRIALS OF THE COSSACK.

ship. In the case of the stern wave the height and distance of the crest astern were accurately measured, and the profile in other respects was sketched by an observer.

The depth of the water in which the runs were made on the Maplins was taken at fairly regular intervals throughout the trials, the course having been previously carefully sounded from end to end, and found to be very uniform in depth.

The data obtained, after correction for any slight variation in displacement, etc., are shown in the figure annexed. It will be seen that the general characteristics of the curves closely correspond to those obtained by Messrs. Yarrow, Herr Popper, Major Rota, Captain Rasmussen, and Messrs. Denny. With her maximum power the *Cossack* is able to develop in shallow water a speed of about 1.4 knots in excess of that which she would develop in deep water at the same displacement and with the same shaft horsepower.

It is probable that the water at Skelmorlie is sufficiently deep to eliminate the effect of the bottom in these experiments. The results given by Major Rota show that a depth of 34 fathoms should be sufficient for this vessel's purpose. This is also confirmed by the experiments with a German torpedo boat in 1904, which showed that at a depth of 40 meters the resistance differed but slightly from that at 60 meters.

The results obtained for the *Cossack* may be used for other vessels of approximately the same size and form, but they do not apply directly to vessels differing materially in size and form.

* From a paper read before the Institution of Naval Architects, April, 1909.

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SOME EXPERIMENTS WITH LIGHTENED BEAM BRACKETS.—II.

BY R. EARLE ANDERSON.

The problem now presented is to embody the results set forth in the previous table in a practical working formula capable of being used in determining the scantlings in any given case. It is evident that any such formula must be wholly empirical, and for this reason, and also for facility in using, it is desirable that the formula be as simple as possible. As the action of the main portion of the bracket is considered to be essentially a strut action, it seems proper to base the formula on some existing column formula in full realization of the facts, however, that the bracket is not a simple strut, and that the constants employed in the various column formulae in common use are based upon tests of a nature wholly different from the nature of the tests undertaken in the present investigation.

After careful scrutiny of the field, the formula most suited to serve as a basis upon which to work appears to be Johnson's parabolic formula, which takes the following form:

$$p = S - c\left(\frac{l}{r}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Where

p is the ultimate unit stress in pounds per square inch,
S is the elastic limit of the material in pounds per square inch,
l is the length of the column in inches,
r is the radius of gyration in inches,
c is a constant.

This formula has the great advantage of simplicity, and involves only one empirical constant. Both of these virtues make it particularly adapted to such use as that to which we propose to put it.

Means for determining the actual elastic limit of the material from which the bracket specimens were made were unfortunately not available. The value of S had therefore to be assumed, and as the nature of the sheet metal and its process of manufacture made it very probable that the elastic limit would be high, the value of S was taken at 40,000 pounds per square inch.

Using this value and taking successively the four average values of p obtained from the four types of brackets, the corresponding values of c are as follows:

Bracket No. 7, $c = 12.45$
" " 8, $c = 13.85$
" " 11, $c = 9.1$
" " 12, $c = 12.45$

These values, except in the case of No. 11, are very close together. The average value is 12. Embodying this value of c in the formula we have

$$p = 40{,}000 - 12\left(\frac{l}{r}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

As was expected, this constant is much greater than those used in Johnson's formula for ordinary columns, the difference being due to the eccentric form of the strut portion of the bracket, and to the eccentricity of the load.

We now pass to the application of this formula to the actual bracket used on the ship. This bracket is shown in Fig. 3 (page 223). The most remarkable way of attacking the problem is to make the bracket capable of sustaining a bending moment equal to, or somewhat greater than, the bending moment which the deck beam or the frame bar is capable of withstanding. In the case of the vessels on which the beam bracket shown in Fig. 3 was used, the deck beam and the frame bar are of the same scantling and their section modulus is 0.724. Assuming that the ultimate strength of the material may be 65,000 pounds per square inch, the greatest possible bending moment which can be withstood by the bar alone is $65{,}000 \times 0.724 = 47{,}000$ inch-pounds.

It is sometimes the practice to consider a portion of the plating to which the beam is attached as acting with the beam, the breadth of plating so taken being generally three times the breadth of the flange of the beam. On this basis, and allowing for a rivet hole, the section modulus is 0.96, with a corresponding ultimate bending moment of $65{,}000 \times 0.96 = 62{,}400$ inch-pounds. In the actual case before us, however, the thickness of the sheer strake and deck stringer plating is very disproportionate to the size of the beam and frame bar, and the connections of plates to bars are by small rivets widely spaced, and hence probably insufficient to make plate and bar act together under any bending moment approaching the ultimate strength of the bar. Moreover, the sheer strake is connected through washer liners which certainly make any such combined action of plate and beam quite impossible. It seems proper, therefore, to take the value derived from the simple beam, namely, 47,000 inch-pounds, as the bending moment to be transmitted by the bracket.

The next step is to calculate the load which would be produced in the two outermost rivets in the bracket by this bending moment. The rivets connecting the deck stringer angle to the plating unquestionably form a part of the bracket connection, and these are taken into account for the extent of one frame space. The area of each rivet is then multiplied by its distance from an assumed axis, and also by the square of its distance, the position of the neutral axis is found by making the statical moment equal to zero, the position of the true axis is corrected for, and the actual moment of inertia of the system of rivets is found, all in the usual manner.

The neutral axis is found to lie 1.795 inches from the center of the rivet in the throat of the bracket, and the moment of inertia is found to be

$$I = 62.54$$

Applying the ordinary formula for uniformly varying stress, namely,

$$p = \frac{My}{I}$$

we have for the stress in the outermost rivet, the distance of which from the neutral axis is 11.205 inches:

$$p_r = \frac{M \times 11.205}{62.54} = 0.179\,M$$

and for the actual load in this rivet, the area of which, when driven, is 0.2485 square inch:

$$P_r = 0.179\,M \times 0.2485 = 0.045\,M$$

Similarly for the next to the outermost rivet we have

$$p'_r = \frac{M \times 7.995}{62.54} = 0.1278\,M$$

and

$$P'_r = 0.1278\,M \times 0.2485 = 0.0375\,M$$

The total direct load passing through the outer or flanged portion of the bracket is thus

$$P_r + P'_r = (0.045 + 0.0375)\,M = 0.0825\,M$$

whence, by substituting the value of M, previously determined, we have for the bracket strut load 3,880 pounds.

The sectional area through the flanged portion of the bracket at midlength is 0.66 square inch. Hence the unit stress, due to the direct load, is

$$\frac{3{,}880}{0.66} = 5{,}880 \text{ pounds per square inch.}$$

To this must be added the stress, due to the bending moment produced by the eccentric loading. The amount of the eccentricity is equal to the distance from the neutral axis of the

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strut section to the heel of the flange, minus the half thickness of the plate, since the load is applied on one side of the plate at one end, and on the opposite side of the plate at the other end. For an 8-pound bracket, the neutral axis is found to be 0.393 inch from the heel, and subtracting the half thickness of the plate, or 0.1 inch, we have for the eccentricity 0.293 inch.

The bending moment, due to the eccentric load of 3,880 pounds is therefore

$$3{,}880 \times 0.293 = 1{,}140$$

inch-pounds. The moment of inertia of the cross-section is found on calculation to be 0.1282 square inch inch square, and as the distance from the neutral axis of the cross-section to the most strained fiber in compression is found in the same calculation to be 0.393 inch, the stress due to the eccentric load is

$$\frac{1{,}140 \times 0.393}{0.1282} = 3{,}480$$

pounds per square inch. Adding the stress due to the direct load to that due to the eccentricity, we have for the total stress in the most strained fiber

$$5{,}880 + 3{,}480 = 9{,}300$$

pounds per square inch.

The next step is to determine whether the bracket is capable of withstanding this stress, and for this we use the formula (2). It appears wise, however, to adopt for actual use a lower value of the elastic limit than was used in interpreting the results of the experiments. The reasons for using the high value of 40,000 pounds for the thin sheet steel experimented upon have been stated. For the ordinary medium steel of which the actual bracket is made, we will use the value of 35,000 pounds as more nearly representing the material to be dealt with. This puts our formula in the following form:

$$p = 35{,}000 - 12\left(\frac{l}{r}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots (3)$$

The radius of gyration of the cross-section, found at the same time as was the position of the neutral axis and the value of the moment of inertia, is 0.441 inch, and as the length of the flange is 20 inches, we have

$$p = 35{,}000 - 12\left(\frac{20}{0.441}\right)^2$$

$$p = 10{,}700$$

pounds per square inch. Comparing this value, which represents the ultimate unit strength of the bracket, with the value previously found as the unit stress to which the bracket may be subjected by a bending moment in the beam sufficient to break the beam, namely, 9,300 pounds per square inch, we see that this bracket has a reasonable, but not too great, margin in its favor, it being highly desirable, of course, that in a case like this, where the fixity of the ends of the beams depends upon the efficiency of their connection, the strength of the connection should be greater than that of the parts connected.

Of course, the strength of the rivets requires also to be looked into. The information for this has already been obtained. In determining the load to be supported by the strut portion of the bracket, the stress in the outermost rivet was found to be

$$p_r = 0.179\ M$$

Putting in the value previously found for the ultimate bending moment of the beam we have

$$p_r = 0.179 \times 47{,}000$$

$$p_r = 8{,}400$$

pounds per square inch, which is, of course, way within the strength of the rivet.

The original design of the torpedo vessels upon which these brackets were used provided for 10-pound plate, but as a result of these experiments this was reduced to 8 pounds, whereby a considerable saving in weight was effected.

Further calculations on brackets of smaller dimensions, but with the same number and size of rivets and of the same thickness of plating, showed that the smaller the bracket the greater would be its strength, due to the shortening of the strut and the fact that the strength of the shortened strut increased more rapidly than did the load transmitted to it by the shortened rivet connection, the limit being reached, of course, when the stress produced in the outermost rivet is equal to the ultimate shearing strength of the material. The large size of the bracket was retained in the design because of the support given by it to the sheer strake and the main deck stringer, quite aside from its ability to resist racking strains. It is important to note, however, as a general principle, applicable to beam brackets, bulkhead stiffener brackets, and other similar structural members, that where the object is simply to resist a given bending moment the smallest bracket that will admit of efficient riveting will, if its edge be properly supported by a flange or by running part of the beam or stiffener out on it, be the strongest.

It may also be noted that where a stiffener of channel bar or I-beam section is split and carried down the edge of the bracket, or where a reverse bar is thus fitted, the stiffening should extend to the toe of the bracket, and it is very desirable where it does not cause too great complication to run the stiffening past the bracket clips.

In regard to the accuracy of the formulas (2) and (3), it should be stated that they are proposed with a considerable degree of reservation, for while the usefulness of the experiments upon which they are based exceeded the writer's expectations, the total number of tests was, of course, not great, and the uncertainty of the elastic limit of the material used introduces an unfortunate element of doubt. Care has been taken, however, to make all necessary assumptions on the safe side, and it is believed that until more elaborate experiments can be made on full-size brackets the formula (3), if used with proper discrimination, will give reasonable and safe results.

Summer Meeting of Naval Architects' Society.

The summer meeting of the American Society of Naval Architects and Marine Engineers was held at Detroit, Mich., June 24 to 26. The program was as follows:

FRIDAY, JUNE 25, 1909.

1. "Some Model Experiments on Suction of Vessels," by Naval Constructor D. W. Taylor, U. S. N., Vice-President.

2. "A Method of Determining Pressure for Steam Turbines," by Professor C. H. Peabody, Member of Council.

3. "The Resistance of Some Full Types of Vessels," by Professor H. C. Sadler, Member of Council.

4. "The U. S. S. *Michigan* renamed the *Wolverine*," by Commander W. P. White, U. S. N., Associate Member.

SATURDAY, JUNE 26, 1909.

5. "Shallow-Draft Steamers," by Charles Ward, Member.

6. "Material Handling Arrangement for Vessels on the Great Lakes," by Alexander E. Brown, Member.

7. "The Strength of Knees and Brackets on Beams and Stiffeners," by H. R. Hunt, Junior Member.

8. "Towing Problems," by T. S. Kemble.

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MAST AND DERRICK MOUNTINGS.

25-TON STEEL DERRICKS.

Fig. 1 shows a 25-ton steel derrick fitted on deck. The derrick is 50 feet long, 22 inches diameter at the center, 18 inches at the ends, and the plating is 7/20 inch thick, with T-bar stiffeners 5 inches by 3 inches by ⅜ inch, with 10/20-inch diaphragm plates. A 10/20-inch doubling plate is fitted in way of the band for a length of 4 feet 3 inches. The band is placed 3 feet from the derrick head. and the doubling is arranged 1 foot 3 inches above the band and 3 feet below it.

The band is 9 inches broad by 1½ inches thick. On top of the band is arranged a snug, 2¼ inches thick, to take the attachment of the topping lift treble block. On the sides of the band 1½-inch wrought eyes are worked with a 1⅜-inch link, the link being made long enough to take the shackle of double guys; on the underside of the band a 2-inch wrought eye is worked to take a 2½-inch link. The link is to take the shackle of the treble purchase block.

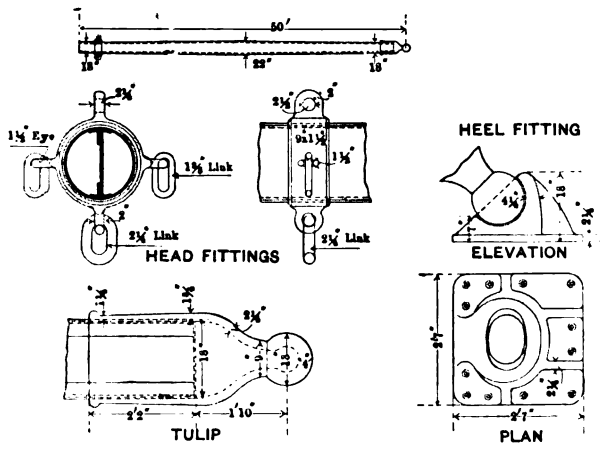

FIG. 1.

The heel socket is the well-known "cup type" of casting. The sole is 2 feet 7 inches broad by 2¼ inches thick, with holes for twelve 1¼-inch bolts. The extreme height is 18 inches. Four brackets are arranged for; one on each side and two on the right-hand side. Each bracket is 2¼ inches thick.

From the center of the ball to the end of the tulip is 4 feet. The diameter of the ball is 13 inches, and it is 4 inches thick after machining. From the center of the ball to the end of the derrick is 1 foot 10 inches. The thickness of casting varies from 4 inches at the ball to 2¼ inches at the end of the derrick. From the end of the derrick to the end of the tulip is 2 feet 2 inches, and the diameter of the derrick for that distance is 18 inches. The thickness of metal in the tulips varies from 1¾ inches at the end of the derrick to 1¼ inches at the end of the tulip. The tulip is 9 inches diameter at the ball. The ends of the derrick are closed with 6/20-inch plates tapped on.

25-TON WOODEN DERRICKS.

Fig. 2 shows a 25-ton derrick 40 feet long, 21 inches diameter at the center and 16½ inches diameter at the ends. This derrick is fitted on a mast, and on the mast are two brackets, 4½ inches deep by 7½ inches broad, with soles 5 inches deep and 1½ inches thick, of sufficient length to take eight rivets on each side of the mast. A distance piece, with jaws, is fitted between the brackets, to take the shod of the derrick. A pin, 3¼ inches diameter, is let down through the brackets and sleeved jaw; 2⅛ inches clear metal is left round the brackets and sleeved jaw. The jaws are 2 inches thick. From the center of the bracket pin to the center of the derrick pin is 9 inches. The snug taking the derrick pin is 7 inches deep by 3¼ inches broad. The length of heel shod from center of derrick pin is 3 feet 10½ inches. The forks of the shod are 4½ inches broad by 1⅛ inches thick, tapering to 3 inches broad by ¾ inch thick. At the heel of the derrick the shod is 1⅜ inches thick. Four 1-inch clinched bolts are fitted. Two bands are fitted 1 foot 10 inches apart, and are 6 inches broad by ¾ inch thick, and 5½ inches broad by ¾ inch thick, respectively. Eyes are fitted on the underside of the bands when it is considered necessary.

The upper band is 1 foot from the head of the derrick, and it is 7 inches broad by 1¼ inches thick; on the upper side of the band a snug is fitted to take a 1¾-inch shackle. This snug is 2 inches thick, and the shackle pin which passes through is 2 inches diameter; the height of the snug is made to suit the shackle. A 1⅜-inch link, taking a 1¼-inch chain as a bridle, is fitted to the shackle; 1¼-inch eyes are fitted at the sides of the band to take a 1⅜-inch link for double guys. On the lower side of the band a 2-inch wrought eye with a 2½-inch hole is worked; through this eye is fitted a link 12 inches by 8 inches by 2¼ inches, to take the shackle of the

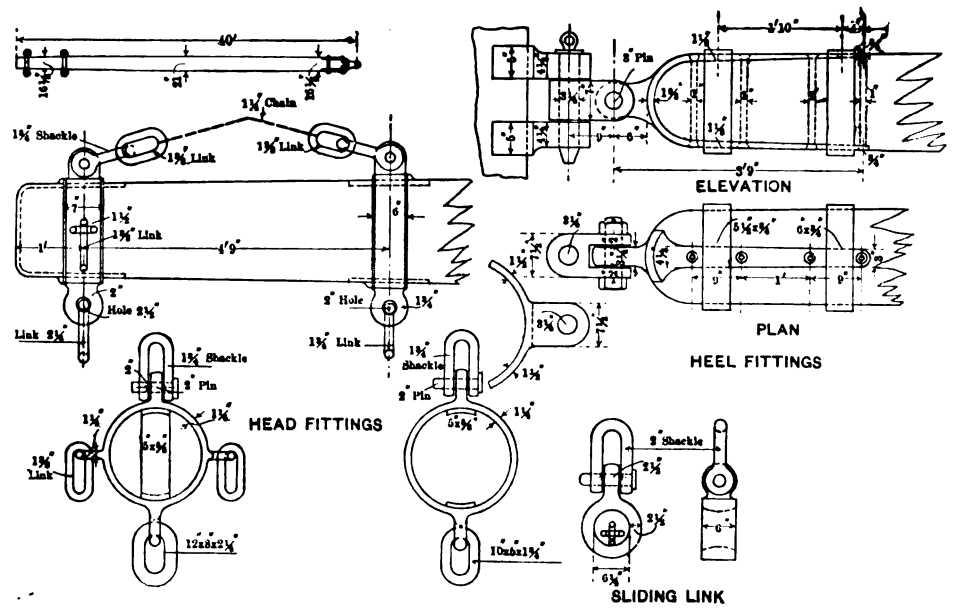

FIG. 2.

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purchase block. At the head of the derrick a plate is fitted, 5 inches broad by 5/8 inch thick, with stops at the top and bottom of the band to prevent the band from moving when loads are applied.

The lower band is spaced from 4 feet 6 inches to 6 feet from the upper band; it is 6 inches broad by 1¼ inches thick, and the snug at the head is the same as for the upper band. No eyes are worked for guys. A 1¾-inch wrought eye is

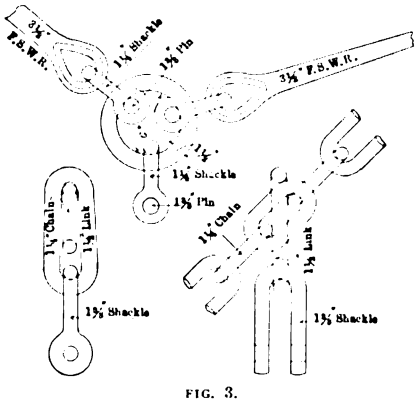

FIG. 3.

worked on the underside, with a 2-inch hole to take a link 10 inches by 5 inches by 1¾ inches, to which is attached the shackle of the purchase block.

The sliding link is 6 inches broad and 2 inches thick. A snug, 2½ inches thick, with height and breadth to suit a 2-inch shackle which is fitted to a sliding link, is also fitted. The 2-inch shackle may either take a 5-inch F. S. W. R. topping lift or a treble block, depending on the requirements of the shipowner.

Fig. 3 shows two types of cargo spans. The upper sketch shows the type usually fitted between masts. The span is 3½ inches F. S. W. R., with a ring 1½ inches thick, having

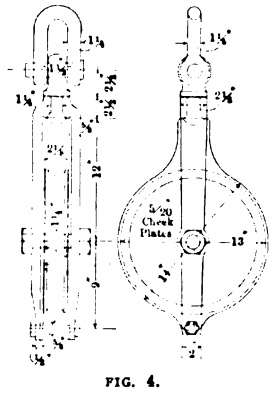

FIG. 4.

an internal diameter of 6 inches. The ring is connected to the span with 1¼-inch shackles. A 1¼-inch shackle is fitted to take a gyn block. This style takes loads of about 6 tons.

The lower sketch shows the method of fitting spans from mast to deck. The length of 1¼-inch chain may vary from a few feet up to 20 feet when the ends are secured to 4¼-inch F. S. W. stays. A long link is fitted, through which is passed a 1⅛-inch shackle, and from this shackle the gyn block is hung. A safe working load on this span would be 10 tons. When it is desired to move the gyn block over any part of the hatch, long links are fitted to eye plates on deck, and the span is shackled to the most convenient one.

The sheave of the lead block is 13 inches diameter and 2 inches thick. The sheave pin is 1¼ inches diameter, grooved for oil and fitted with a feather at the head to prevent the bolt turning. From the center of the sheave to the underside of the crown is 12 inches, and from the center of the sheave to the center of the distance piece is 9 inches. The crown is 2½ inches deep. From the crown to the center of the shackle pin is 2½ inches. The swivel head is 1½ inches thick, with a breadth and height to suit a 1¼-inch shackle; the swivel is 1¼ inches diameter. The jaws of the block at the crown are 2½ inches broad by 5/8 inch thick, and at the distance piece 2 inches diameter by ½ inch thick. The check plates are 5/20-inch thick, and the over-all width of the block is 14 inches.

PROPELLER COMPUTATIONS.

BY CHARLES S. LINCH.

In the issue of INTERNATIONAL MARINE ENGINEERING for February, 1905, there appeared in the editorial on propellers the following:

"The propeller of the present day is very largely an evolution. This evolution has proceeded, not along strictly scientific lines, but, if we may so state it, along the lines of least resistance. One designer has followed blindly in the footsteps of another, giving vent, perhaps, to a few of his own ideas in the matter, but being in the main fully as mindful of precedent and subservient to it as would cheer the heart of the most pettifogging lawyer to be found in a day's journey.

"The existence of such conditions would appear to indicate that there is no scientific basis for the design of the screw propeller, which conclusions are by no means in accordance with the truth."

The truth of the above remarks are self-evident to any one conversant with the methods of design carried on in some of our shipyards. There seems to be an abhorrence of scientific investigation, and an utter disregard for the compilation of data, preference being manifest for rule-of-thumb methods and so-called judgment. I do not mean to say this obtains in all cases, but so much of it has come under the observation of the writer that at times one wonders how it is possible that such conditions obtain. It is true that one or two of the principal works are considered too mathematical, but to those who care to design a wheel from formulæ derived from scientific investigation these works can be used. Durand's *Resistance and Propulsion of Ships* is so arranged that one needs but ordinary arithmetical knowledge to solve the problem of the proper proportions. Taylor's *Resistance of Ships* presents no difficulties, and the student or designer who cannot handle these works in their entirety should not be entrusted with the design of a propeller wheel, or, in fact, any marine machinery, as the days of rule-of-thumb methods are fast passing away.

It should be the aim of every designer to have at hand a series of curves or characteristics of blades—and in a short time one can accumulate quite a number which will cover a very wide range. The idea of considering the time spent in plotting these curves or making the necessary computations as wasted, or that they are too difficult for every-day use, is an utter fallacy.

In the issues of INTERNATIONAL MARINE ENGINEERING for December, 1907, and January, 1908, there was published a very complete description of the Hamburg-American steamer *Kronprinzessin Cecilie*. We will take this ship as an example, and not only show how closely both Durand's and Taylor's formulæ agree, but we will lay down the wheel and show the characteristic curves, and substitute the values and solve the equations.

DATA.

The designed indicated horsepower of each engine is 3,035; the designed revolutions per minute 79. The sea speed is taken at 14 knots.

From indicator cards in the writer's possession of a similar engine the friction was 10 percent. For a ship of these lines we would assign a wake factor of about 15 percent, a true slip of 25 percent, and the apparent slip will be 14 percent.

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COMPUTATIONS BY DURAND'S METHOD.

The useful work is equal to (pitch × revolutions)^3 × (diameter)^2 × constants; or, if written as a formula, we have

$$U = (P \times R)^3 \times D'^2 \times i \times k \times l \times m.$$

The values of the constants are given in Durand's work. The writer enlarged the curves of area ratio and thrust ratio m from Durand's analysis and from Froude's investigations, and the values are more accurately determined.

PITCH.

14 × (1 — wake factor) = 14 × (1 — 0.15) = 11.9 knots.

11.9 × 101.3 = 1,205.47 feet per minute.

(1 — 0.25) $P \times R$ = 1,205.47,

or 0.75 $P \times R$ = 1,205.47 feet

Therefore, $P \times R$ = 1,607.29,

But R = 79 per minute.

Therefore, P = 1,607.29 ÷ 79 = 20.345 feet.

The pitch as built was 20.343 feet.

DIAMETER.

The log of $P \times R$, or 1,607.29 ÷ 100 = log of 16.0729 = 1.206099.

Log (16.0729)^3 = 1.206099 × 3 = 3.618297.

Corresponding number = 4,151.6.

The value of k = 0.193.

The value of l = 0.670.

The value of m = 1.05.

The value of i = 1.03.

Assume an efficiency of 63 percent.

Therefore, useful work = 3,035 × 0.9 × 0.65 = 1,720.845 horsepower.

Now, substituting and solving for D, we have

$$D'^2 = \frac{1{,}720.845}{4{,}151.6 \times 1.03 \times 0.193 \times 0.67 \times 1.05} = 2.96$$

D^2 = 2.96 × 100 = 296, $D = \sqrt{296}$ = 17.2 feet.

Therefore, diameter of wheel = 17.2 feet.

The value of i is the blade factor.

The diameter of wheel, as built, was 17.0625 feet, or a difference of 1.65 inches, or 0.8 percent.

DEVELOPED AREA.

Taking the thrust ratio, or m, as 1.05 and passing over to where, by judgment, the curve of pitch ratio, 1.18, would intersect the horizontal, we find the area ratio would be 37 percent. That is, the area of developed surface would be 37 percent of disc area. The disc area would be 232.35 square feet; therefore, the area of developed surface would be 85.969 square feet. The area as given was 85.23 square feet, a difference of 0.739 square feet, or 0.134 square feet for each blade. The diameter as built being 17.0625 feet, the corresponding disc area is 230.465 square feet, and 37 percent of 230.465 square feet equals 85.27 square feet.

What is involved in this computation? A designer should have at hand wake factors for different ships and also blade factors, and when equipped with these he is in a position to handle these equations with a degree of confidence.

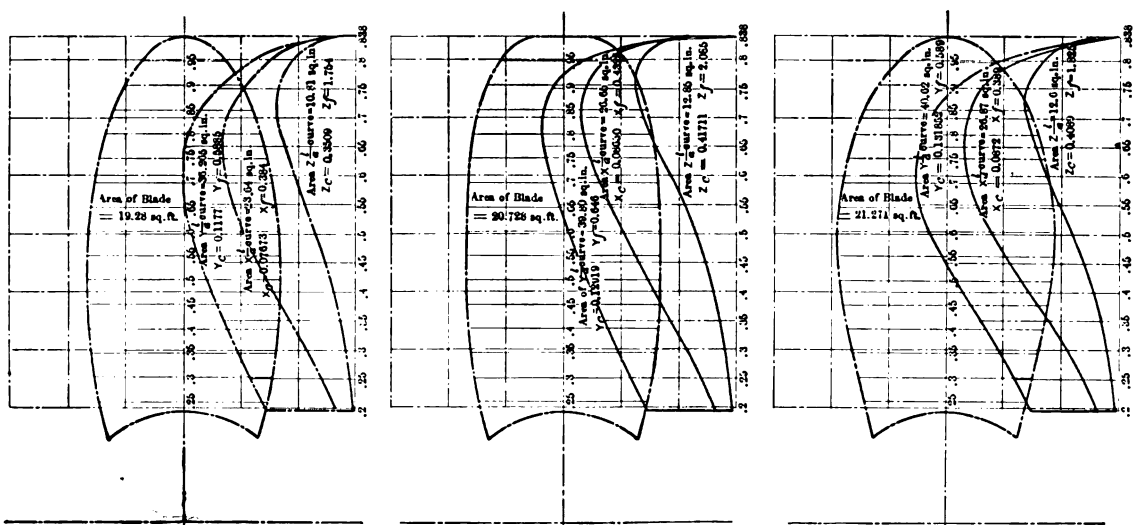
COMPUTATION BY TAYLOR'S METHOD.

It is not out of place to say right here that Mr. Taylor's methods were rigorously deduced from the experiments of Froude, and are based upon his investigations. In the figure is shown the blade as built, and the three curves determining the characteristics X_t, Y_t and Z_t. Now, the brake-horsepower is equal to three times the number of blades in one wheel times the square of the diameter and constants, or

$$B.\ H.\ P. = 3\ Z \times \left(\frac{P \times R}{1{,}000}\right)^3 \times D^3 \times b\ (aS_t X_t + fZ_t).$$

Where Z = Number of blades, one wheel.
D = Diameter in feet.
P = Pitch in feet.
R = Revolutions per minute.
b = Mean width ratio.
a = 8.4 — 1 × diameter ratio.
S_t = True slip.
X_t = Characteristic.
f = Coefficient friction = 0.045.
Z_t = Characteristic.

Let us take the diameter as built, namely, 17.0625 feet:
We found $P \times R$ = 1,607.29.



STANDARD ELLIPTICAL BLADE DIAM. 17.0625 FT. PITCH 20.343 FT.		STANDARD BLADE DIAM. 17.0625 FT. PITCH 20.343 FT.		BLADE AS BUILT DIAM. 17.0625 FT. PITCH 20.343 FT.	
Mean Width Ratio	= 0.171	Mean Width Ratio	= 0.184	Mean Width Ratio	= 0.189
X_f For this Ratio	= 0.448	X_f For this Ratio	= 0.470	X_f For this Ratio	= 0.461
Y_f " " "	= 0.668	Y_f " " "	= 0.702	Y_f " " "	= 0.698
Z_f " " "	= 2.052	Z_f " " "	= 2.267	Z_f " " "	= 2.164

COMPARISON OF DIFFERENT TYPE BLADES.

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$$\frac{P \times R}{1{,}000} = 1.60729.$$

Log 1.60729 = 0.206099.
Log $(1.60729)^3$ = 0.618297.
Corresponding number = 4.1516.
Diameter = 17.0625.
Log 17.0625 = 1.231979.
Log $(17.0625)^2$ = 2.463958.
Corresponding number = 291.1.

$$\text{Diameter ratio} = \frac{17.0625}{20.345} = 0.838.$$

The value of $a = 8.4 - 0.838 = 7.56$.
The value of $Xt = 0.461$.
The value of $Zt = 2.164$.
Substituting the values in the above equation we have

$$B.\ H.\ P. = 3{,}035 \times 0.9 = 2{,}731.5.$$

Therefore, solving for mean width ratio we have

$$b = \frac{2{,}731.5}{3 \times 4 \times 4.15 \times 291.1 \times (7.56 \times 0.25 \times 0.461 + 0.045 \times 2.164)} = 0.19$$

But $b = \frac{\text{mean width}}{\text{diameter}}$, therefore mean width = 0.19 × 17.0625 = 3.24 feet.

$$\text{Mean width} = \frac{\text{area of blade}}{\frac{\text{diam. wheel} - \text{diam. hub}}{2}}$$

Therefore, area of blade = 3.24 × 6.544 = 21.202 square feet.

The developed area of four blades equals 21.202 multiplied by 4 or 84.808 square feet. The diameter of hub was made 3.9739 feet. The area of Fig. 1 by planimeter was 21.271 square feet. This would make the total area of the wheel equal 85.084 square feet. The difference is due probably to the difference between points taken for area by the builder and the writer. In looking over the two methods we find for Durand's method the diameter of wheel is less than 1 percent larger than built. The developed area is nearly 1 percent greater. The pitch is exactly as built. With Taylor's method we would use the diameter as built. The pitch would be exactly as built, and the developed area 0.1 percent less than built.

In plotting curves of every blade we soon have a series of curves, the value of which to the designer cannot be overestimated.

Sufficient has been said to show the value and the readiness of computations from these investigations, and in analyzing the performance of wheels the ease and readiness with which they can be handled commend themselves to the designers. The time spent in using them is far less expensive than replacing new wheels on a ship, though it is hard to convince some of this fact.

Results with Producer-Gas Motor Boat.

The producer gas motor boat *Marenging*, recently built for H. L. Aldrich, publisher of INTERNATIONAL MARINE ENGINEERING, and described on page 110 of our March issue, has now been in service nearly two months, and has traveled approximately 1,248 miles. Anthracite pea coal has been used on all these runs, and the results, exclusive of dock trials, show that the boat will average between 800 and 900 miles on a ton of anthracite pea coal.

BREAKDOWNS AT SEA.

Repairing a Broken Stern Gland at Sea.

A few years ago, while I was fourth engineer of the steamship *V*———, en route to San Francisco and the Hawaiian Islands, we encountered that well-known and treacherous weather which frequents the region of the Straits of Magellan during the months of June, July and August. As the wind and sea grew more and more violent the ship rolled and pitched excessively, the indicator in the engine room at times showing a list of as much as 35 degrees. Notwithstanding the severe weather, however, everything in the engine room was working satisfactorily, with the exception of the thrust and

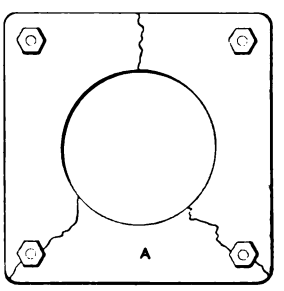

FIG. 1.

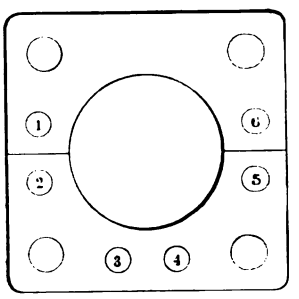

FIG. 2.

spring bearings, all of which carried a very high working heat.

Things continued in this way for the first two days, but on the third day, when the storm was at its height and everything on board the ship was getting a thorough sea test for its stability, one of the oilers, while paying his half-hourly visit up the trembling shaft alley, discovered that the port stern gland had been broken by the excessive springing of the tail shaft. The nature of the fracture is shown in Fig. 1, the piece *A* having dropped out and allowed some of the flax packing to work partly out at that point.

In order to effect repairs all hands were called and the port engine stopped, as there was a very dangerous shaft

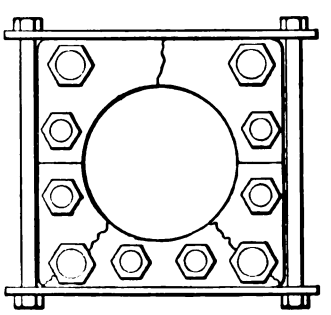
FIG. 3.

coupling revolving about 18 inches from the broken gland. After the engine was stopped it was found that there was no loose, heavy iron or steel with which to make suitable repairs strong enough to withstand the vibrating strains which had caused the fracture. Nevertheless, we had to do something, so the chief engineer gave orders to unhinge a small, heavy iron door about 2 feet by 5 feet by ½ inch thick, that led from the donkey boiler room to the 'tween decks. This was marked off the exact size of the gland, Fig. 2, and then it was cut in two, and a 2-inch hole drilled in each corner for the studs to pass through; also 1½-inch clearance holes were drilled at 1, 2, 3, 4, 5 and 6 to correspond with the repair studs that were being put into the broken pieces of the gland by two oilers.

After the door had been cut up, making two complete plates

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with a total thickness of 1 inch, it was necessary to place a strip around the plates to clamp the four sides of the gland together; so, with the balance of the iron door two pieces were made 2 feet 8 inches long and 6 inches wide, with a 1½-inch hole drilled in each end. Then a slice bar was cut up, making two studs, 2 feet 6 inches long, which passed through the holes and clamped around the gland, as shown in Fig. 3. After bolting on the double thickness of plates and setting up the gland, the engine was started and the ship proceeded on its voyage.

This repair job held out remarkably well, considering the fact that the bad weather kept up for a week longer, and no further trouble was experienced with it during the remainder of the trip, which occupied about six months, four months of which was running time.

The Use of Wood for Breakdowns on Board Ship.

It is not suggested that wood should form the basis of a permanent repair, but that it forms a very useful medium where the ship does not carry a sufficient number of spare parts to enable the engine room staff to do more than patch up the job until the ship arrives at port.

It frequently happens that when the circulating pump bucket is packed with rope the chamber becomes so much scored that the working life of even the best rope that can be used for packing is only a few days. In order to get over this wear of

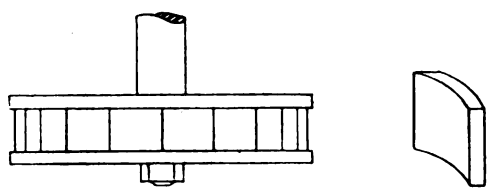

FIG. 1.

material, and the constant annoyance of stopping, due to these causes, a good plan is to pack the bucket with wood in the following manner: The staves of a barrel which has been used for packing pork or beef or oil may be taken and cut off to the required length in order to fit between the faces of the bucket, as shown in Fig. 1. They should then be fitted into the bucket, using a rough file or rasp to make them of suitable shape. The diameter should be slightly slack, in order to allow the wood to swell under the action of the water. It will be found that this will make a good, tight job, and also that the action of the wood will in almost every case be such as to smooth down the scores on the interior surface of the chamber, so as to enable rope to be used again in the pump.

A common mishap on board ship is the fracture of the rams on the feed or bilge pump, and should there not be a suitable spare part a wooden ram may be fitted, as shown in Fig. 2.

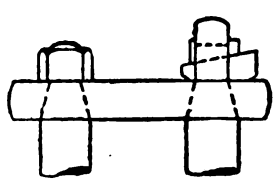

FIG. 2.

In such temporary repair work a boat's oar has been used for this purpose, making the wood a good fit in the cross-head. Soft packing should be used and the gland left slack until the wooden ram has expanded. This is a valuable method in case one feed pump ram is not able to adequately feed the boilers.

A rather daring method of repair with wood is in packing the steam piston. It sometimes happens that the piston itself fractures, as shown in Fig. 3, a certain amount of metal being left on the piston rod, the rest breaking away. As a temporary repair wood may be used, as shown in Fig. 4. Wood segments are made so as to fit into the metal remaining on the

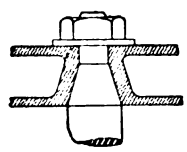

FIG. 3.

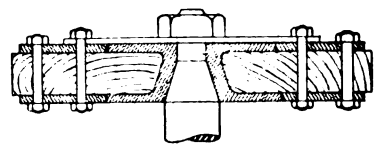

FIG. 4.

piston rod, and these are bound together with iron plates on either side, fitting up to the fracture and clamped by means of bolts, in order to further strengthen the part. The washer may be replaced by a piece of steel plate extending over the fractured part, having holes drilled in it to receive the bolts which clamp the plates on the upper and under side together. In order to preserve the strength of the wood it may be useful to make the packing up of two or three layers of wood, so arranged that the grain of one layer crosses that of the other; also a split-joint arrangement could be adopted. This not only insures steam tightness, but also gives greater mechanical strength in order to resist the bending action due to the steam pressure.

Repairing a Broken Circulator Piston.

In September, 1906, I was filling the position of third assistant engineer on the steamship *R*————, bound from New York to Galveston, Tex., and the first day out we encountered extremely rough weather, which at times caused us to slow the engine down owing to the excessive pitching and rolling of the vessel. Things went well at first, except that the thrust bearing carried a high heat, due to the sudden thrusts put upon it every time the ship would pitch and the engine give those well-known quick jumps before being throttled. On the fourth day, however, at 1.30 A. M., when the second assistant was on watch, a sudden succession of rapid cracks from behind the condenser suddenly brought the sleepy oiler to his senses, and, just as he ran around the engine, he saw the circulating engine's piston rod, cross-head brasses, etc., break loose from the connecting rod and

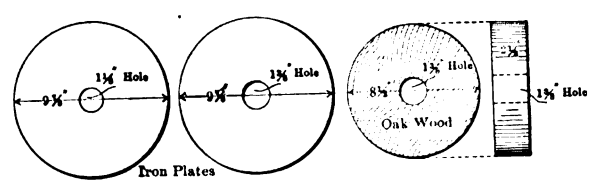

FIG. 1.

pound up and down a few times in the cylinder. With the circulator broken down and no pumps available to act in its stead, the engine had to be stopped until all hands could get below and rig up a jet condenser, the necessary connections being fortunately available for this purpose.

After the engine was again started, the cylinder head of the circulator was taken out, and it was found that the piston was completely smashed into small pieces beyond the hope of repair. The chief gave orders to make a temporary piston out of wood, the exact size of the cylinder bore, and reinforced with iron plates. This resulted in an argument between the chief and the first assistant, who claimed that a piston made in this way would swell in a short time and grip the cylinder walls and stick fast, whereas if it were made of a smaller diameter to allow for swelling it would permit too

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much steam to leak by to operate the engine. The chief finally had one made the exact size of the cylinder bore, reinforced with circular plates cut from an ashpan damper, which were placed each side of the wooden piston and fastened with through bolts. After this was finished the engine was started up, but it ran only about five minutes when it stuck hard and fast, so that it was necessary to split it up to get it out.

We then proceeded to make another piston along more mechanical lines. We first cut two circular plates from another ashpan damper ⅛ inch smaller than the diameter of the cylinder bore, the top plate with a center hole 1⅛ inches diameter and the bottom plate with a center hole 1⅜ inches diameter to correspond with the taper on the piston rod.

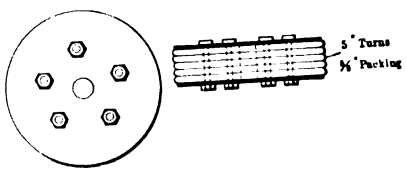

FIG. 2.—FINISHED PISTON.

Then a circular piece of oak wood, 2½ inches thick, with a 1⅜-inch hole through the center, was cut out, 1½ inches less than the diameter of the bore. After the center holes in the two plates and the wooden disc were bored they were bolted tightly together, while five ½-inch holes were drilled through the piston to receive binding bolts. The piston was then bolted securely together and placed in the cylinder, and fitted on the piston rod. While placing in the cylinder, however, five turns of ⅝-inch rod packing were wound around the wooden part. This junk piston was tested by blocking the fly-wheel and admitting full steam pressure on top with the bottom drains open, allowing the steam to leak by the piston for a few minutes until it was fully expanded and became steam-tight. The engine was then started, and the piston gave satisfactory service for two months before a new cast iron one was made. F. J. W.

How to Run an Engine With a Broken High-Pressure Slide Rod.

A good many marine engineers do a lot of unnecessary work in order to get their engines into running order after the breaking of their high-pressure slide rod, an accident which is not at all uncommon at sea. They go to the extent of disconnecting the high-pressure connecting rod, slinging up that engine, blocking up the steam ports and various other miscellaneous jobs, before they get their boat under way again. It may therefore be of interest to briefly detail a method that was adopted in the breaking of a high-pressure slide rod in a two-cylinder compound engine. The spindle was broken in the valve at a bad weld, and all that was necessary to run the ship into port on the low-pressure engine was the following procedure:

The high-pressure slide valve was taken completely out, and the rod was examined in order to see if it was bent in the stuffingbox. As this was not the case it was left in its place, free to move up and down with the motion of the eccentric. Liners were placed under the feet of the low-pressure valve rod in order to obtain an additional amount of steam on the up-stroke, and the steam pressure was reduced in order to suit the engine working on the low-pressure cylinder only. After one or two attempts the low-pressure crank passed over its top center, and after this everything was plain sailing. As the high-pressure piston was in equilibrium, there being equal pressure on the top and bottom surfaces, there was no necessity at all to block up the steam ports, and there was also no necessity for disconnecting the connecting rod and slinging up the high-pressure engine. The extra steam consumed in pulling round this engine did not amount to very much, although, of course, the consumption of coal was slightly increased. A great point, however, was that the time taken in effecting the repair was considerably reduced over the ordinary method above described, and this is a very important factor where the vessel may be in a heavy sea.

The Fracture of a Tail Shaft.

On board a boat which was proceeding under water ballast, with a quantity of sand ballast in the afterhold, a fracture occurred in the tail shaft just inside the stern tube, as indicated in the sketch. The first step in order to make repairs was to remove the sand ballast from the afterhold onto the forward deck and also to pump out the water ballast in the after-hold. This set the ship down by the head, thus lifting the stern tube clear out of the water. At this stage of the proceedings it was found that the propeller and the afterpart of the shaft had dropped off, and were, of course, lost.

It was next attempted to draw the shaft through the liner into the boat, but it was found that this could not be done, and jacks had to be applied in order to push the crank shaft out again. A staging was therefore hung over the stern of the boat, and upon inspection it was found that the shaft had broken at a point about 3 inches inside the liner, and it had expanded this liner, thus preventing the shaft from being drawn through because of a cutting action. It was therefore necessary to cut off the expanded part; and after this was done the shaft was got into the boat. A wooden block was then made and put in the stern tube until the spare shaft was made ready. Fortunately the ship was well equipped for the job, and there was a spare shaft and propeller in reserve. Moreover, the after end of the tunnel was built to such a height that the broken shaft could be taken out and the new one put in without having to take the tunnel top off. Bars were fitted onto the tunnel and bolted onto the same, upon which the shaft rested, so that there was no necessity to use blocks of wood. At the same time the whole experience was a rather trying job, because, as the ship was in the Atlantic, it was constantly rolling, which rather impeded the progress of the work.

A Broken Slide-Spindle Block.

Fig. 5 shows the construction of a slide-spindle block which was fitted on a boat trading between Great Britain and America. In order to take up the wear and tear of the block on the top and bottom faces of the link, and to render it adjustable at all times, a fitting called the "gib key" is fixed to the block by means of two ½-inch tap bolts on each side. It is customary to reduce this adjustable fitting a sufficient amount to allow of several thin liners being placed on each side, in order to take up the great wear and tear occasioned by the constant working of the block on the go-ahead end of the link.

In the case under consideration this adjustment had been made a few days previous to the accident, and, unfortunately, it had been left too tight, so that when the boat was stopped for a pilot and the engines were ordered "full astern," the block suddenly snapped off at the top as shown in Figs. 1 and 2. This practically disabled the engine, and as the next step of the voyage was to proceed up the Mississippi, where a considerable amount of "backing and filling" would take place, it was considered more advisable to come to anchor and make an immediate sound repair of the fracture than go ahead with the partially disabled boat. Fortunately, a small country blacksmith's shop was found in the vicinity of the anchorage,

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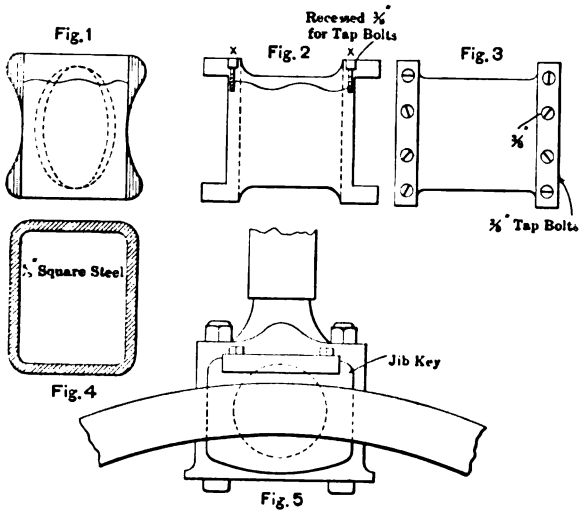
and he had a few pieces of square section steel bar of various sizes which would form raw material for the repair.

The first step was to chip off the rounded ends of the block, as shown in Fig. 1, in order that a good bedding surface could be obtained for the bands which were to be applied. Four holes were then drilled and tapped on each side of the block, well down into the lower half, as shown in Fig. 2. Each hole was ⅜-inch tapping size, and the holes were recessed sufficiently to allow the tap bolts to be let in flush with the top of the block. In order to bring these bolts squarely into place, slots were placed in their heads, and they were firmly and tightly screwed into position by means of a screwdriver fitted into a ratchet brace. The appearance of the repaired piece is shown in Fig. 3.

A steel band of ⅝-inch square section was then made of bar material, strongly welded, and this was firmly shrunk over the four sides of the block at the part previously occupied by the



rounded shaded portions in Fig. 1, one of these bands being placed at each end of the block. This effectually covered up the recessed heads of the tapped bolts and rendered them safe from working slack and coming out. The whole repair was a very neat and substantial job, and—from another vital standpoint—it was a very cheap one. The blacksmith's charge for his part of the work, including labor and material, was only $6, and the engine-room time occupied was only a few hours. In order to show that it was a perfectly sound repair it may be mentioned that the boat came through a very fierce Atlantic gale with it, and had it in use for nearly four months before it was possible to get it replaced by a new one.

It was decided by the inspecting engineer to keep this repaired block as a standby, and at the same time it was suggested that the three go-astern ends of the links should be softened and reduced in order to render the links capable of being changed end for end when necessity arose. It may be of value to suggest here that if the majority of engine builders were to adopt the construction already made by one or two well-known builders, and were to fit the pin in the center of the link where the drag arms are coupled, the tedious process of softening and reducing could be entirely obviated.

Broken Coupling Bolts on a Marine Shaft.

In the event of the bolts in a shaft coupling working slack and breaking away, if some form of replacement is not made the ship is virtually helpless, and it frequently occurs that there are no spare parts for such a purpose on board ship. This has been overcome in practice by taking out one bolt from each of the other shaft couplings and fitting these into the coupling which is disconnected by the broken bolts, and the repair has been strong enough to bring the ship home.

THE MARINE STEAM ENGINE INDICATOR.—I

BY LIEUT. CHARLES S. ROOT, U. S. R. C. S.

Almost coincident with his invention of the steam engine, James Watt brought out the steam engine indicator. Hand in hand they have progressed towards that goal of all inventions—perfection. Improvements in the steam engine have been met with equal advancements in the design of the indicator to adapt it to such new conditions as have arisen. Volumes have been written on both subjects, and at the present time there is little that is new to be said on either. However, of the many treatises on the indicator, there is none which does not contain some feature not embodied in other books on the subject, and the object of the writer in the following chapters will be to bring out the best features of the standard literature on the subject, and in general to give a resumé of marine indicator practice as it exists to-day.

HISTORICAL.

The original Watt indicator consisted of a small steam cylinder, open to the atmosphere at one end and in communication with the engine clearance space at the other, through a suitable pipe and cock. Working in the cylinder with a nice

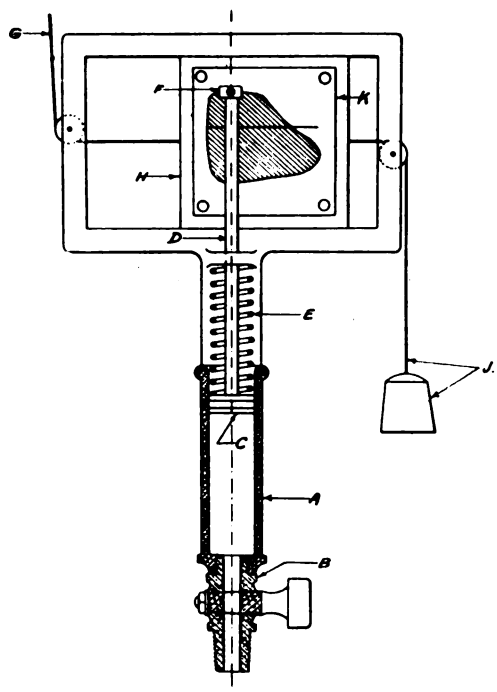


FIG. 1.

sliding fit was a small piston, its rod towards the outer end of the cylinder, the combination being constrained to move in a straight line by a guide near the outer end of the rod. The movement of the piston was regulated by a helical spring surrounding the rod, fixed to the frame of the instrument at one end and to the piston at the other. When the pressure in the cylinder was greater than that of the atmosphere the spring was compressed, and when the exterior pressure exceeded that within, the spring was extended. With the engine in operation the indicator piston was caused to reciprocate, the changing pressures being indicated by a pointer attached to the outer end of the piston rod sliding over a fixed scale. With the slow, rotative speeds of the early engines, it was possible to obtain an idea of the action of the steam in the main cylinder by watching the pointer, and Watt made great use of the instrument in this shape in perfecting his engine.

The first improvement on the original instrument was made by Watt himself, and consisted of the addition of a sliding

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panel moved by the parallel motion of the main engine. The instrument as used about 1815 is shown in Fig. 1. *A* is the steam cylinder, *B* the stop cock, through which connection is made with the engine cylinder, *C* the piston, *D* the piston rod, *E* the spring, *F* the pencil holder, *G* the cord attached to the parallel motion of the engine for the purpose of pulling the panel *H* to the left, and *J* the cord and weight for hauling the panel to the right on the return stroke. With the instrument connected up and the engine in operation, the panel with the paper card (*K*) attached, had a reciprocating motion horizontally, which was a reduced copy of the motion of the engine piston, and its position at any instant was an index

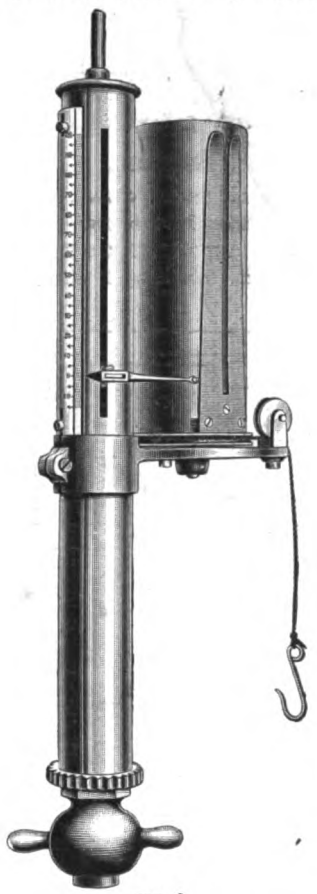

FIG. 2.

of the piston position at that time. The pencil, by its vertical height, indicated approximately the varying pressure in the engine cylinder at every instant. The combination of these two motions caused the pencil to draw a closed diagram on the card as shown in the figure. The inclosed area has been cross-hatched in order that it may be easily distinguished.

McNaught's indicator, which followed that of Watt, is shown in Fig. 2. This instrument was in general use until about 1862. It differed from the Watt instrument in that a drum, turning on a vertical spindle, was fitted in lieu of the sliding panel, and a spiral or helical spring inside the drum took the place of the counterweight. Various forms of this instrument were made in Great Britain and the United States. Fig. 2 was taken direct from an early American McNaught indicator, now in possession of the American Steam Gauge & Valve Manufacturing Company, of Boston, Mass.

Owing to the length of spring and the long piston stroke necessary to produce legible cards, this instrument was suitable only for the most moderate pressures and rotative speeds. When an attempt was made to use this apparatus with the higher speeds coming into use in the fifties, the long and tremulous spring was put into violent oscillations by the momentum of the moving parts, with the result that the cards were neither legible nor trustworthy. As an example of the best that could be done with the McNaught indicator at speeds of about 200 revolutions per minute and 130 pounds of steam, we quote from Mr. Charles T. Porter:*

"The two preceding diagrams (Figs. 3 and 4) . . . are fair average samples of a large number taken in February, 1856, by the late Daniel Kinear Clark, from the locomotive *Canute*, on the London & Southwestern Railway, with an indicator of the best construction then known, and which had been expressly prepared for the purpose. . . . The attempt to conjecture what the true form of these diagrams should be—to learn from them, for example, what proportion of the boiler pressure was obtained in the cylinder, and how much the pressure fell before reaching the point of cut-off at the speed of piston employed—points which it is of the highest consequence to ascertain—is clearly hopeless. It is to be observed, also, that the pencil does not, in either case, follow the same line during the successive revolutions of the engine, but describes quite different lines."

When such results as Figs. 3 and 4 were the best that could be obtained by a highly-skilled operator with a specially constructed instrument, it can easily be seen why the indicator was fast falling into disrepute as an instrument of precision,

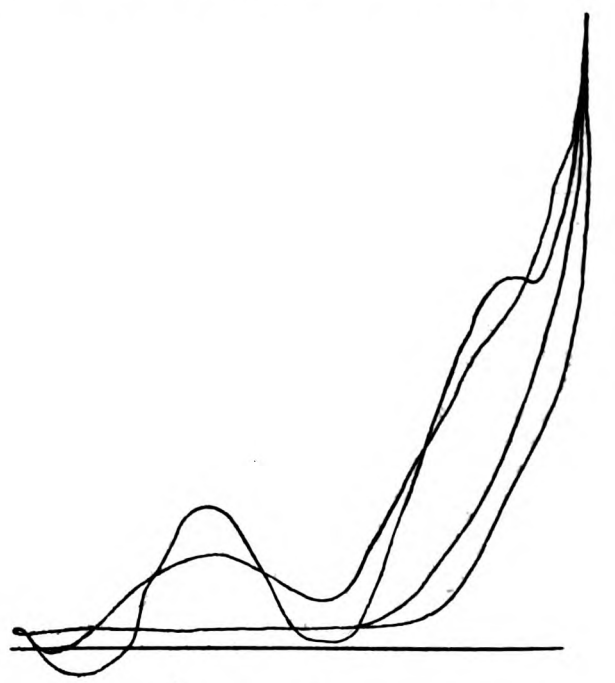

FIG. 3.

and had not an improvement been made about this time the indicator must have gone out of use, except for slow speeds and low pressures.

About 1862, as a result of efforts to improve the instrument, Mr. C. P. Richards, of Hartford, Conn., brought out the indicator bearing his name, which embodied nearly every essential feature found in modern types. The pencil was connected to the end of a lever of the third order, the piston being joined near the fulcrum. This arrangement reduced the travel of the indicator piston and the length, compression and extension of the spring. Thus the errors due to inertia and the long spring of the Watt and McNaught instruments were much reduced. The so-called multiplying parallel motion used on this instrument—by means of which the pencil was constrained to move in an approximately straight line—was an invention of Watt. He used this linkage to guide the crossheads of his engines in lieu of the now familiar cross-head

* "The Richards Steam Engine Indicator."

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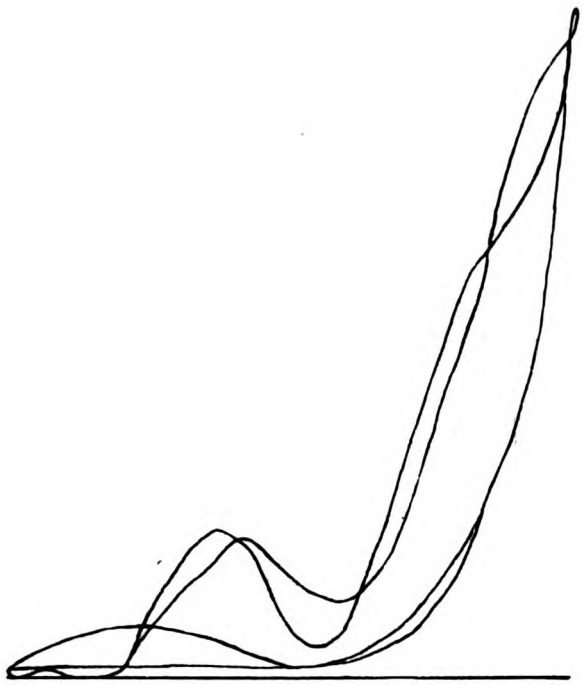

FIG. 4.

slides; but to Richards must be given the entire credit of applying it to the indicator. This motion will be discussed in detail hereafter. A drawing of the Richards indicator, copied from an old cut, is shown in Fig. 5.

The continual increase in speeds and pressures made desirable an instrument with a lighter pencil mechanism than that of Richards, and to meet this want Mr. J. W. Thompson brought out his instrument about 1875. Fig. 6 shows one of the first of this pattern made in the United States. The drawing was made directly from an indicator which had been in use at sea for upwards of twenty-five years. In the Thompson instrument, with its light pencil mechanism, we have the essentially modern design. The original layout has been changed but little, and at the present time it is still considered to be one of the best forms for general use.

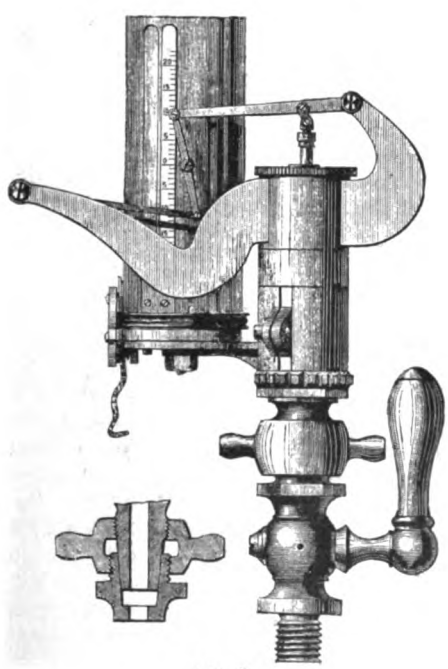

FIG. 5.

MODERN INSTRUMENTS.

The writer will endeavor to describe here some of the important details of the modern indicator, more especially those which differ most in the various instruments, leaving to the readers' good mechanical judgment the selection of a design which will best serve his purpose. For ordinary valve setting the most *extreme* accuracy may not always be necessary, just as it is unnecessary to use a micrometer for taking the data from which to compute the pitch of a screw propeller or to use the ship's chronometer when reading the engine counter for revolutions per minute. In most instances, however, it is best to use the greatest care in the selection and manipulation of the instrument, especially where high rotative speeds or other difficult conditions must be met. In taking data to be used in subsequent designs (to which use cards turned over to the engineer superintendent are liable to be put); or where premiums are paid or penalties exacted for horsepower on trials, or where changes are to be made involving the expenditure of money, the best instruments are not good enough. Generally speaking, the sea-going engineer is advised to obtain the best instruments possible, attach them to his engines correctly and manipulate them with the greatest precision.

In the perfect instrument, the pencil should show by its vertical height the pressure beneath the indicator piston at

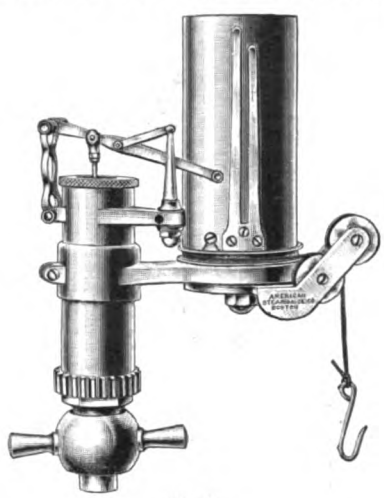

FIG. 6.

any instant, and by its horizontal position on the diagram the position of the engine piston at the corresponding time. Simple as these conditions are it is impossible to fulfill them, owing to the fact that the various parts of the instrument must necessarily have weight, and therefore inertia. Inertia may be defined as that property of matter by which it tends, when at rest, to remain so, and when in motion to continue in motion in a straight line, unless acted upon by some external force. Generally speaking, the effects of inertia increase with the weight or mass of a body and the speed with which it moves. It is therefore evident that the best instrument is the one in which the least weight is moved through the least distance in the production of diagrams of equal area.

STEAM CYLINDERS.

The steam cylinders in most indicators are fitted with liners very similar in principle to those used in the cylinders of marine engines. They are usually made of a bronze alloy suited to the varying temperature to which the indicator is exposed.*

The general idea is expressed in Fig. 7. The liner is centered and secured at *A*, and as it is out of contact at all other parts it is free to expand and contract longitudinally. With

* As ammonia attacks all of the bronzes, indicators are entirely steel fitted when used on ammonia compressors.

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this arrangement no lateral distortion occurs with change of length; worn liners are easily renewed by the insertion of duplicates, and the cylinder area is easily reduced—when desired—by fitting liners of smaller bore. The annular space *B* fills with steam of the same pressure and temperature as that beneath the indicator piston and acts as a sort of steam jacket.

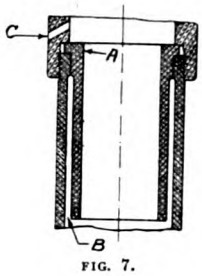


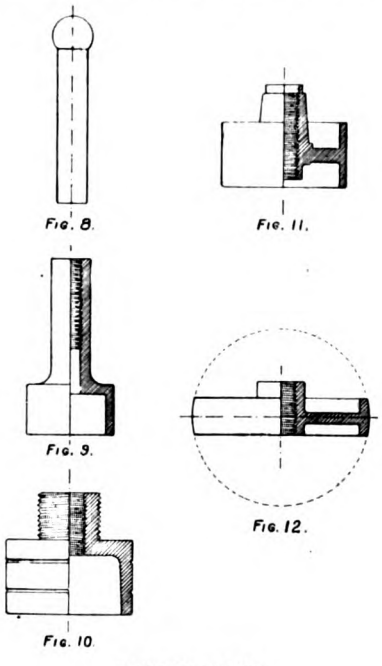
FIG. 7.

The outlet *C* prevents back pressure or the formation of a partial vacuum above the piston.

The bore areas of these liners vary from 1 square inch—in some of the newer steam-engine instruments—to 1/32 inch, or even smaller, in those used for heavy ordnance and hydraulic work. The areas usually decrease in a regular geometric series with a ratio of ½, thus 1, ½, ¼. * * *

PISTONS.

In the design of indicator pistons, the principal points aimed at are lightness, reasonable steam tightness and freedom of movement. They are made as light as possible consistent with strength, and the various makers have found by



TYPES OF PISTONS.

experience the limit to which they can go in this direction. Steam tightness is sought in most cases by making the pistons quite deep, and in some instances by turning grooves in the working surfaces for water packing. Only approximate tightness is necessary, however, as the supply pipe is large, and unless the leakage of the piston is so great that a back pressure is created on the atmospheric side it will affect the instrument but little. Lack of freedom of movement is usually caused by the piston becoming "cock-billed" in the cylinder. This may be caused by the spring buckling or the piston rod being out of line, and is usually guarded against by connecting the piston rod—and sometimes the spring also—to the piston by means of some form of universal joint.

A few of the various types of pistons are shown in Figs. 8 to 12. Fig. 8 is a form used in ordance and hydraulic work where heavy pressures are met with, and has a comparatively small area. Fig. 9 is a ¼-inch area piston, used for gas-engine work in a steam-engine indicator with a bushed or reduced cylinder. Figs. 10 and 11 are common forms of ½-inch area pistons used for steam-engine work only. Fig. 12 is a 1-inch area piston, whose working surface is an equatorial zone of a sphere. With this form the piston cannot jam, even when canted. Pistons are made of both bronze and steel. Diaphragms similar to those used in reducing valves have been used in lieu of pistons, and in one form of instrument the steam gage spring tube is used.

PISTON SPRINGS.

In order to obtain a correct diagram, the motion of the piston must be exactly proportional to the pressure beneath the indicator piston, and if the spring is not correct the entire instrument is useless, so far as a correct measurement of

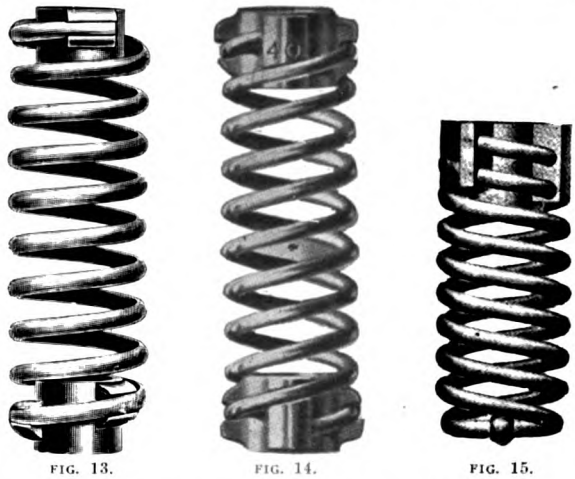


FIG. 13. FIG. 14. FIG. 15.

power is concerned. The piston spring is, therefore, one of the most important single details of the instrument.

Springs are usually rated by the number of pounds pressure which, acting on a ½-inch area piston, will give the pencil a movement of 1 inch, and this quantity is known as the scale of the spring. The following scales are most generally used: 6, 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 56, 60, 64, 70, 72, 80, 100, 120, 150 and 200. Some engineers prefer to use scales which are even multiples of ten, while others prefer scales which correspond with the divisions of the ordinary rule, as 8, 16, 32. * * * Indicator manufacturers can also furnish instruments whose springs are adjusted to the metric system. When pressures higher than 250 pounds are met with the piston area is usually reduced. Thus with a ¼-inch area piston a 100-pound spring will give the same result as would a 200-pound spring used with a ½-inch piston.

The majority of indicators are fitted with helical piston springs. The following exceptions may be noted, each one representing a class: The Batchelder, having a flat spring; the Keynion, a steam-gage tube, and the Hädike a flexible diaphragm. The two latter have no pistons, but the Hädike is equipped with an auxiliary helical spring.

The helical springs are made of the finest steel wire, tempered by the most experienced workmen. Those made of a single wire are usually wound in a single thread on mandrels, with from four to four and one-half threads per inch, while the double, or so-called "duplex," springs are wound in a double thread with a pitch of about two "turns" per inch.

Fig. 13 shows a much-used form of single spring, and Figs. 14 and 15 double or duplex springs. The springs are generally made a little stronger than necessary, and reduced to the standard by grinding or by screwing them into or out of the

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bronze spring heads. It will be noticed that the spring shown in Fig. 15 has but one head, and though wound in a double thread it is made of but a single piece of wire. The bead at the bottom acts as the center member of a ball and socket joint, by which the spring is attached to the piston. This permits of great freedom of movement of the piston relative to the center line of the spring. All springs, of whatever form, are carefully calibrated by the makers before being issued. The process of calibration will be described later. The spring heads are generally threaded on the inside for attachment to the instrument.

(To be continued.)

THE NEW HUDSON RIVER STEAMER ROBERT FULTON.

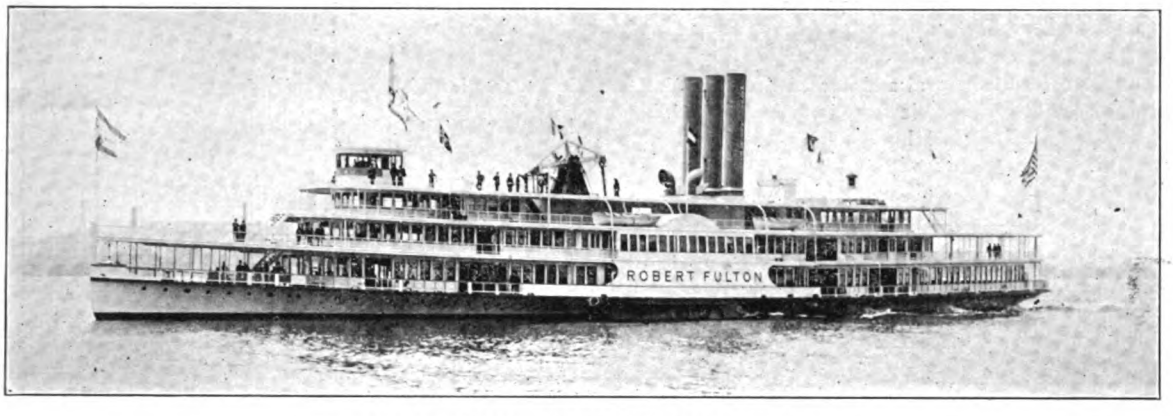
The latest addition to the famous fleet of Hudson River steamboats is the *Robert Fulton* of the Hudson River Day Line, the principal dimensions of which are as follows:

Length between perpendiculars	336 feet 0 inches
Length over all	346 feet 0 inches
Breadth of hull, molded	42 feet 0 inches
Breadth over guards, molded	76 feet 0 inches
Depth, base line to top of deck beams at side of hull at lowest point of sheer	12 feet 4 inches
Crown of deck beams in 76 feet	12 inches
Gross tonnage	2,168
Net tonnage	1,344

The conditions governing the construction of the *Robert Fulton* were unusual, since, not only is she a boat of remarkable design, involving many new features, but also on account of the short time available for her construction it was necessary to use the utmost dispatch in carrying out every feature of the work. The design of the boat was placed in the hands of Frank E. Kirby, D. E., consulting engineer, and J. W. Millard, naval architect. The hull and joiner work were constructed by the New York Shipbuilding Company, Camden, N. J. The machinery and boilers were built by W. & A. Fletcher, Hoboken, N. J., while the interior decoration of the boat was in the hands of Mr. Louis O. Keil.

The hull is of steel and is divided into five watertight compartments by four transverse bulkheads. The frames are of bulb angles, 4 inches by 3½ inches by 8 pounds, spaced 24 inches apart. As this size bar is not rolled in the United States, it was necessary to send to Scotland for the frames. These were ordered even before the contract for building the hull had been let, and it is noteworthy that the material was delivered to the builders within twenty-four hours of the time it was needed. The floors are of flanged steel 12½-pound plate, 15 inches deep, and the main-deck beams are bulb angles, 4 inches by 2½ inches by 8 pounds, placed on every frame. The sheer strake of plating is 17½ pounds by 45 inches wide amidships, gradually tapering to 10 pounds at the bow and stern. In way of the wheel opening the sheer strake is increased to 20 pounds. The bottom and side plating are 12½ pounds amidships, reduced to 10 pounds bow and stern. The landing edges of all outside strakes, except the garboard and sheer strakes, are joggled, doing away with the use of liners.

Throughout the superstructure of the boat the construction is entirely fireproof, steel, asbestos and other non-combustible materials being used exclusively. The strength and rigidity of the structure are secured by means of a system of stanchions between the decks. These are of steel placed in four rows extending practically the whole length of the vessel. By means of connections to the longitudinal girders and deck beams, the entire structure is thoroughly braced to withstand the hogging and sagging stresses set up by the rapid shifting of the load on the boat, which consists almost entirely of passengers, it being necessary to make provision so that the entire complement of 2,000 or more passengers can all be placed either at one end or at one side of the boat, as may be necessary on account of the direction of the wind or position of the sun. This fact also had its effect upon the design of the hull, which must have good stability under all these conditions. This led to the placing of large trimming tanks on either guard just aft of the paddle wheels. These tanks have a capacity of 20 tons of water each, and there is also a ballast tank holding 30 tons of water located in the hold at the stern of the boat.



THE NEW HUDSON RIVER STEAMER ROBERT FULTON.

The design of the hull was further complicated by the fact that on a draft of less than 7 feet it was necessary to build a hull of sufficiently fine model, so that a speed of 23 miles an hour could be obtained without the expenditure of an extraordinary amount of power.

The boat is propelled by steel feathering paddle wheels, 30 feet 8 inches diameter at the outside of the rims. The buckets, or paddles, are each 12 feet 6 inches long by 3 feet 6 inches wide. The engine is of the single-cylinder, vertical, surface-condensing walking beam type commonly used on American river steamers. The cylinder is 75 inches in diameter, with a stroke of 12 feet, and runs at about 40 revolutions a minute. The normal speed of the boat is 20 miles an hour, but, under favorable conditions, she can be forced to 23 miles an hour. Steam is supplied by three lobster-back return tubular boilers, each 33 feet long, operating at about 55 pounds pressure per square inch. The total heating surface of the boilers is 5,499 square feet, and the total grate area 228 square feet, making a ratio of heating surface to grate area of 24.1. The

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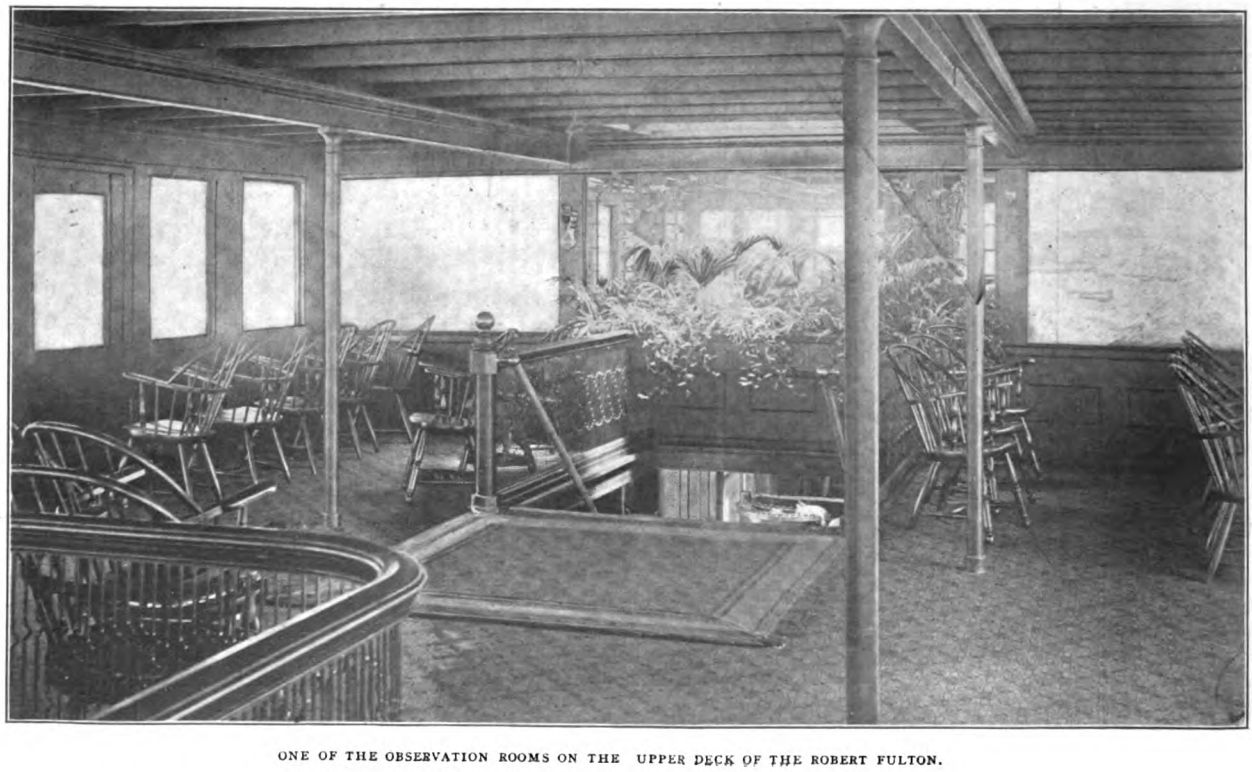
GRAND SALOON OF THE ROBERT FULTON, DESIGNED AS AN ITALIAN GARDEN.

boilers are fitted for both natural and forced draft, the latter being supplied by two Sirocco blowers, each having a double inlet and single outlet. The wheels are each 42 inches in diameter in full casing, so designed as to give a total air delivery of 56,000 cubic feet of air per minute against a pressure of 2½ inches of water, the fans operating at 475 revolutions per minute. Each fan is direct connected to an 8½-inch by 6-inch double-inclosed engine, designed to operate on an initial pressure of 35 pounds per square inch with a high vacuum.

The necessary pumps, ventilating fans and electric generating apparatus are all located on the starboard side of the



ONE OF THE OBSERVATION ROOMS ON THE UPPER DECK OF THE ROBERT FULTON.

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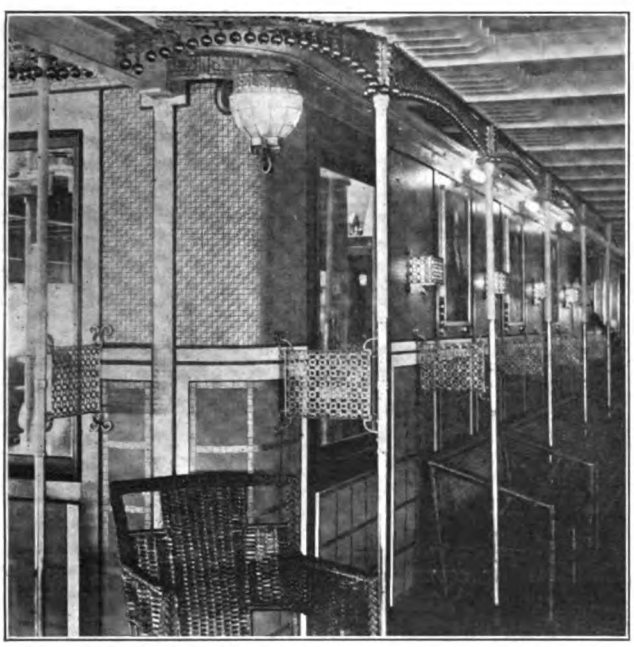
engine room. One of the electric generating sets is driven by a Curtis turbine. The donkey boiler is located on the main deck. An elaborate automatic fire alarm system has been installed, including three annunciators, placed in the pilot house, engine room and the purser's office. The alarm is given simultaneously on these three annunciators and by means of bells in the crew's quarters, so that in the event of fire or any other emergency the crew will immediately respond, knowing the exact location of the trouble.

The general arrangement and interior decoration of the boat are both novel and pleasing. The forward part of the hold is utilized for the crew, and aft of this is a lunch room



IRON GRILL WORK IS A FEATURE OF THE INTERIOR DECORATION OF THE ROBERT FULTON.

70 feet long, extending the whole width of the ship. This room is fitted up in ship-cabin style, the dining tables being set against the sides of the hull, each in a separate compartment, with wood benches on each side. The wood used for the furniture and wainscoting is cypress. The combination of the quaint design with the open ports and deck beams overhead gives a very pleasing effect. Just aft of the lunch room is the engine room, and aft of the engine room the boiler room. Separated from the boiler room by a bulkhead is the galley, which is about 34 feet long and contains a complete outfit of ranges and galley accessories, besides a complete refrigerating system.

The forward part of the main deck is entirely inclosed by a deck extension above, while aft are located the hospital, barber shop, toilets, coat and baggage rooms, purser's office, and at the stern of the boat a large dining room, the decoration of which is carried out in Delft blue.

The grand saloon, which occupies the entire saloon deck, is a decided novelty in naval architecture. Instead of the usual style of decoration employed on most sound and river steamers, the saloon is treated as a formal Italian garden with carved pillars, broad balustrades, wide open spaces and irregular nooks, the entire garden being decorated with plants, palms, vines, even canary birds being added to give a touch of outdoor life. Large comfortable wicker chairs are placed throughout the garden. The color scheme of the saloon is white, green and gold. On this deck are placed oil portraits of notable Americans by Robert Fulton Ludlow, great grandson of Robert Fulton, the first marine engineer, and paintings of the old Livingston manor house, of Fulton's first steamboat, the *Clermont,* and the departure of the *Clermont* on her first trip in 1807.

Other paintings include pictures of old cities and towns of the early nineteenth century by Vernon Howe Bailey and Frederick W. Glover, a series of historical paintings showing the development of the steamboat and the various types of sail and other craft that plied on the Hudson in the early days by S. Ward Stanton, and two large decorated wood panels by Raphael A. Weed, depicting Rip Van Winkle's meeting with Hendrick Hudson and Robert Fulton at Clermont. The use of decorative iron work, designed by H. O. Schmidt, also adds much to the beauty of the boat.

The writing room is located amidships, while at the forward end of the saloon deck provision is made for an orchestra on a sunken platform open at the sides, so that the music can be heard on all three decks. Located along the sides, amidships, are ten day-parlors, the walls being decorated with flowers of the Hudson Valley. Each room has a private balcony overlooking the water.

The upper deck includes two large observation rooms, finished in cypress. Large plate-glass windows on three sides of each room give an unobstructed view in every direction.

The keel of the vessel was laid on January 12, and she was launched March 20, the honor of christening the vessel being given to Miss Anita Merle-Smith. The boat had her first trial trip on May 8, sailed for New York on May 20, and went into service on May 29.

A Practical Comparison of the Advantages of Higher Cylinder Ratios.

BY LIEUTENANT C. S. ROOT.

An examination of the cylinder diameters of naval engines will show cylinder ratios varying from 1:4.75 to 1:11.2 in triple-expansion engines of ships of war now in commission. This seems to point to a decided difference of opinion among naval engineers in regard to these ratios, and for this reason it is thought that the following account of two moderately long runs of the same vessel with different cylinder ratios may be of interest.

Passed Assistant Engineer E. T. Warburton, U. S. N., in the *Journal of the American Society of Naval Engineers* (Vol. IX.), describes a trans-Atlantic run of the United States steamer *Bancroft,* from which the following is taken:

"The *Bancroft* was docked and refitted in September (1896) at the navy yard, New York, in ten days, for a cruise on the European station. * * * The *Bancroft* left Tompkinsville, Staten Island, N. Y., Sept. 15, 1896, using both boilers, and arrived at Fayal, Azores, Sept. 25. The dynamo was run about twelve hours, and about 300 gallons of water distilled every day. The vessel started with about 160 tons of bituminous coal on board. Distance steamed, 2,133 nautical miles; time, 10 days 1 hour, or 10.04 days; average speed, 8.85 knots; average revolutions per minute, 124; coal used for all purposes, 102.69 tons; coal per day, 10.22 tons; miles per ton of coal, 20.77. There remained in the bunkers 57.5 tons.

"Left Fayal Sept. 28 and arrived at Gibraltar Oct. 4. Strong head winds were encountered for the greater part of the distance. Distance steamed, 1,136 nautical miles; time, 6 days 3.75 hours, or 6.16 days; average speed, 7.69 knots; average revolutions per minute, 129.2; coal used for all purposes, 59.33 tons; coal per day, 9.63 tons; miles per ton of coal, 19.14.

"Left Gibraltar Oct. 6, and arrived at Smyrna, Asia Minor, Oct. 15. Distance steamed, 1,631 nautical miles.

"The coal obtained at Gibraltar was of such very inferior quality that the daily consumption necessarily increased.

"The trip from Tompkinsville to Smyrna (4,900 nautical

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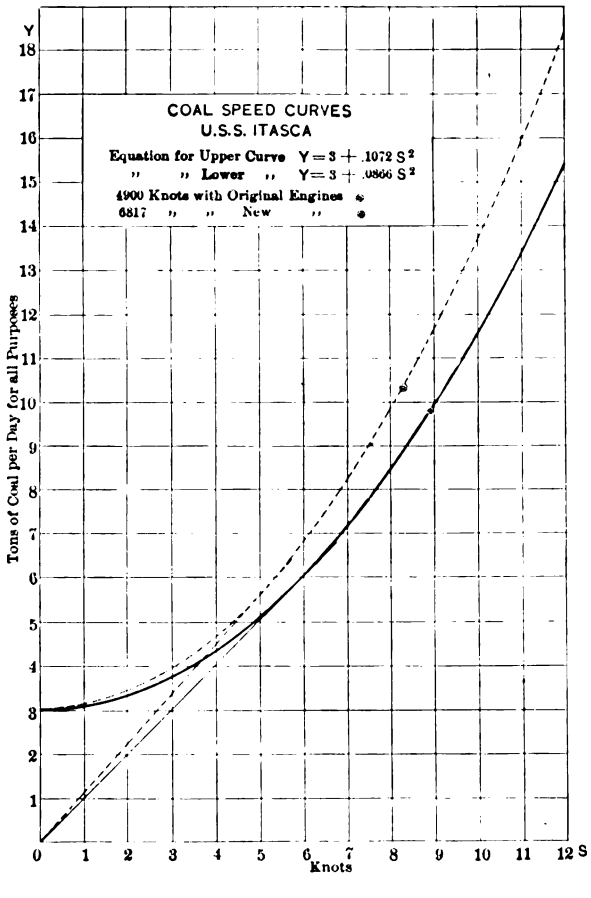
miles) in 29.5 days, including stoppage of 2½ days at Fayal and 2 days at Gibraltar, at an average speed of 8.25 knots and an average daily coal consumption of 10.3 tons, is not a bad showing for this little vessel. * * *"

In the run described the vessel was equipped with two gunboat or low-cylindrical boilers, having 2-inch tubes. The total heating surface was 2,686 square feet, and the grate surface 87.75 square feet. Her twin-screw, triple-expansion engines had cylinders 13½, 21 and 31 inches diameter and 20 inches stroke; the maximum designed working pressure being 160 pounds per square inch above the atmosphere.

In January, 1906, this vessel was transferred to the United States revenue cutter service, and her name changed to *Itasca*. During the following winter she was given a thorough overhauling and partial rebuilding. Watertube boilers of the Babcock & Wilcox type, with 2-inch tubes, were installed.


COAL SPEED CURVES
U.S.S. ITASCA
Equation for Upper Curve Y = 3 + .1072 S²
" " Lower " Y = 3 + .0866 S²
4900 Knots with Original Engines
6817 " " New " "
Tons of Coal per Day for all Purposes
Knots
Y
S

The boilers have a total heating surface of 3,825 square feet and a grate surface of 85 square feet. The matter of cylinder ratios was carefully gone over. New cylinders were designed to give a maximum card factor and minimum terminal pressure while maintaining the same referred mean effective pressure that obtained in the old arrangement. The most suitable diameters were found to be 12 and 19 inches for the high-pressure and medium-pressure cylinders. The low-pressure cylinders were not changed. The "main-engine" auxiliaries, which included an independent, twin-cylinder, single-acting air pump, remained as before, but a larger electric generating set was installed, as was also a larger fan for forced draft and ventilation. A feed-water heater was added to her equipment, and the maximum steam pressure raised to 215 pounds per square inch above the atmosphere. The propellers remained as before, and the vessel was ballasted to bring her down to her original displacement.

After the trial trips the vessel was taken to the revenue cutter service yard at Arundel Cove, Md., and hurriedly equipped for sea. She left there on July 20, 1907, and called at New York, N. Y., and New London, Conn. She coaled at the latter place, and, leaving there on July 28, arrived at Ponta Delgada, Azores, Aug. 7. The data obtained on this part of the voyage were unreliable, for reasons which it is not necessary to enumerate here, and these data will not be included in the figures used hereafter. The *Itasca* left Ponta Delgada Aug. 9, called at Gibraltar, Marseilles, Naples, Algiers, Funchal (Maderia), and arrived at St. Thomas, Danish West Indies, Sept. 27. Total running time, 765.6 hours (31.9 days); total distance steamed, 6,817 nautical miles; total coal consumed for all purposes, 314.5 tons; average steam pressure, 152 pounds per square inch. An average of 700 gallons of water was distilled every day. The coal obtained at Algiers was of the poorest quality, and it was difficult, at times, to maintain the low speed fixed for the voyage. This coal lasted for upwards of 2,000 miles.

The *Bancroft* had strong head winds from the Azores to Gibraltar, and the *Itasca* encountered a gale in the Gulf of Lyons. On each voyage the vessel was handicapped by bad coal for portions of the run. While the *Itasca* distilled more water per day than the *Bancroft*, this was more than balanced by the feed heater. The weather during both runs was uniformly good, except as noted above.

Here we have two long runs at low speeds with practically no change in the vessel, except in the high-pressure and medium-pressure cylinders of the main engines and the addition of a heater, for which a definite allowance can be made. The conditions for comparison seem to be almost as favorable as could be desired. To sum up the two voyages, we have:

	Bancroft.	*Itasca.*
Ratio of the net piston areas	1 : 2.4 : 5.3	1 : 2.5 : 6.8
Grate surface, square feet	87.75	85.0
Time out of drydock at beginning of voyage, months	0	2
Total distance, nautical miles	4,900	6,817
Average speed, knots	8.25	8.9
Average coal consumption, tons per day	10.3	9.86
Water distilled, gallons per day	300	700
Dynamo in operation, hours per day	12	12

In an article published in Vol. XV. of the *Journal of the American Society of Naval Engineers*, Lieut. D. S. Mahony, United States navy, has shown that, if the relation between coal consumption and speed follows any law it is probably as follows:

If rectangular co-ordinates be used, speed in knots plotted as abscissæ and coal consumption in tons per day as ordinates, the curve thus found will differ very little, if at all, from a curve satisfying the equation $y = c + kx^3$. From data in Lieut. Mahony's article is deduced the fact that the actual plotted values of the coal-speed curves of a large number of vessels of the United States navy did not vary, on an average, from the form $y = c + kx^3$ by more than 3 percent, the maximum variation being 6.4.

This form of curve will, therefore, be used in making the comparison.

Let y = the coal consumption in tons per day for all purposes at the speed s.

c = the coal consumption in tons per day for all purposes at zero speed, *i. e.*, with all the usual auxiliaries in operation and the engines kept well "warmed up," making, say, 400 revolutions per hour.

k = a constant which must be computed for each vessel.

s = the speed of the vessel in knots.

Then will $y = c + ks^3$.

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In the case of the 4,900-mile run of the *Bancroft* we have: $y = 10.3$, $c = 3$, $s = 8.25$. Hence, $y = 3 + .1072\ s^2$. For the 6,817-mile run of the *Itasca*: $y = 9.86$, $c = 3$, $s = 8.9$. Therefore, $y = 3 + .0866\ s^2$.

The value of c was taken from the naval records. These curves have been plotted and are shown on the accompanying diagram. The difference in coal consumption is seen to be in favor of the later arrangement of machinery, and at 9 knots is equal to $(3 + .1072 \times 9^2) - (3 + .0866 \times 9^2) = 1.67$ tons per day.

The feed heater raised the temperature of the feed 80 degrees F., and, taking the evaporation as 8 pounds of water per pound of coal—which seems ample in view of the quality of coal used throughout the voyage—a simple calculation will show that about one-third of the 1.67 tons of coal saved per day was due to the heater. The remainder of the saving cannot be accounted for unless it be credited to the superior economy of the new cylinder ratio of the propelling engines.

The following facts in regard to this particular steam plant are brought out by the above comparison and the trial data of the vessel:

1. The engines with the larger cylinder ratios developed higher power on a slightly smaller grate surface.
2. They have shown greater economy at low speeds.
3. The weight of the engines is a little less, due to the reduction of the high-pressure and medium-pressure cylinder diameters.

The substantial increase in economy, due to a small increase in cylinder ratios and steam pressures, is again illustrated in the case of the United States steamers *Newport* and *Annapolis*. Their hulls are similar, as shown below, both vessels being composite and sheathed with copper:

	Newport.	*Annapolis.*
Length between perpendiculars, feet	167.75	168.0
Beam, molded, feet	36	36
Mean draft, feet	12	12
Displacement, tons	1,010	1,017
Area of immersed 'midship section, square feet	354	357
Block coefficient	.482	.49
'Midship section coefficient	.82	.82
Load waterline coefficient	.743	.74

Their engines are as follows:

Newport—Jacketed: Cylinders, diameter, 13½, 23½, 36 inches; stroke of pistons, 30 inches; ratio of high pressure to low pressure by net piston areas, 1 : 5 : 67.

Annapolis—Not jacketed: Cylinders, diameter, 15, 24½, 40 inches; stroke of pistons, 28 inches; ratio of high pressure to low pressure by net piston areas, 1 : 7 : 27.

The following data are from cards taken on the official trials:

	Newport.	*Annapolis.*
Pressure at boilers, pounds per square inch, gage	177	224
M. E. P. referred to L. P. cylinders	42.8	46.58
Piston speed, feet per minute	684.25	686
Indicated horsepower	904	1,217
Terminal pressure of the P. V. curve from combined diagram	20.3	18.63
Water per I. H. P. per hour	15.39	13.33

The superior efficiency of the machinery of the *Annapolis*, as shown by the water rate above, was afterwards maintained in service. Another point indicated by these data is, that had the *Annapolis* been fitted with a 36-inch low-pressure cylinder, the cylinder ratios remaining as before, they would have developed power equal to the *Newport* and must have weighed less. These results check with the runs of the *Bancroft* and *Itasca* described above.

Data, from runs made at low speeds, and selected because of similarity of conditions, are given below:

	Newport. July, 1908.	*Annapolis.* Aug. & Sept., 1908.
Months out of drydock	7	16
Fore and aft sail set, percent of time	50	50
Bunker capacity, tons	232	222
Full-load displacement, tons	1,128	1,116
Duration of the run, hours	156	257.3
Speed, in knots	8.1	8.2
Nautical miles, per ton coal	17.4	26.97
Endurance, days	19	30.4
Nautical miles	3,722	5,987

S. S. GEORGE WASHINGTON.

The latest addition to the transatlantic fleet of the North German Lloyd Line is the *George Washington*, of 27,000 gross tons, built by the Stettiner Maschinenbau-Actien-Gesellschaft "Vulcan," Stettin-Bredow. She is the largest vessel ever built in Germany, the principal dimensions being as follows:

Length over all	722 feet 5 inches
Length between perpendiculars	697 feet 6¼ inches
Length on water line	700 feet 0 inches
Beam	78 feet 0 inches
Depth from upper saloon deck	54 feet 0 inches
Depth from awning deck	80 feet 0 inches
Draft	33 feet 0 inches
Displacement at above draft	36,000 tons
Horsepower	20,000
Speed	18.5 knots

The *George Washington* is a first-class twin-screw passenger and freight steamship, with a straight stem, semi-elliptical stern, two funnels and four pole masts. The hull is of steel, built to the highest class of Germanischer Lloyds, and has a double bottom extending throughout the entire length of the ship. Twelve watertight bulkheads subdivide the ship into thirteen watertight compartments. Eight of these bulkheads extend up to the upper deck and four to the lower promenade deck. The Stone-Lloyd system of watertight doors is used in all the watertight bulkheads, there being 36 doors in all.

The vessel is propelled by twin screws, 21.33 feet diameter and 24.93 feet pitch. The blades, of which there are four on each propeller, are adjustable and can be arranged to give an increased pitch up to 26.58 feet. Each propeller is driven by a four-cylinder quadruple expansion engine having cylinders 38.19, 56.69, 79.92 and 112.21 inches in diameter, with a stroke of 66.93 inches. At 86 revolutions per minute, using steam at 220 pounds per square inch, the engines develop approximately 10,000 horsepower each, driving the ship at a speed of 18.5 knots. On her trial trip a speed of 20 knots was attained, the engines developing nearly 22,000 horsepower. The arrangement of cylinders in the engine is, from forward aft: high-pressure; second-intermediate; low-pressure; first-intermediate. The high-pressure and first-intermediate cylinders each have a single piston valve, the second-intermediate has two piston valves, and the low-pressure cylinder a flat slide valve. The diameter of the crank shaft is 22.25 inches, the tunnel shafting, 20.39 inches, and the tail shaft 22.09 inches. Each engine has a separate surface condenser, having 12,930 square feet of cooling surface. The air pumps are 10 by 32.99 inches by 20.98 inches stroke, and the circulating pumps 12.21 by 51.18 inches by 9.84 inches stroke.

Steam is furnished by eight double-ended and four single-ended Scotch boilers, placed in two separate fire-rooms. All

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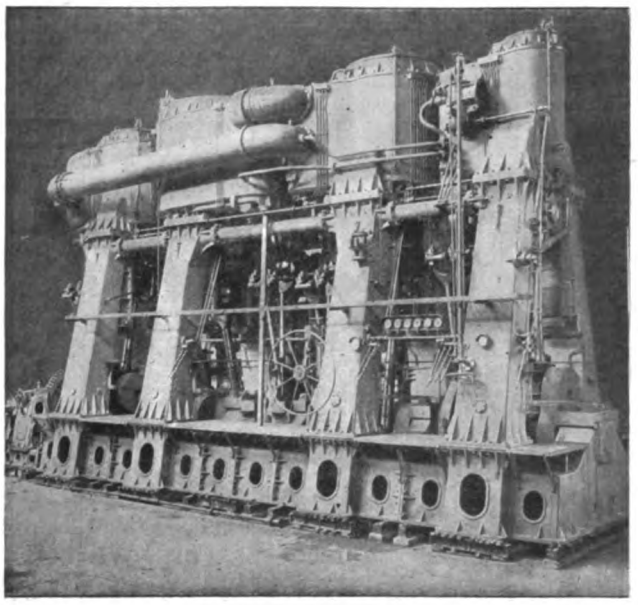
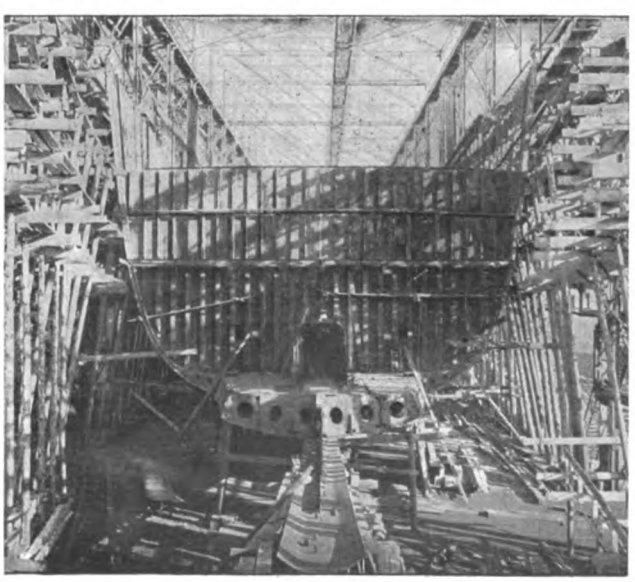


ONE OF THE MAIN ENGINES.

the boilers are 15.75 feet in diameter, the double-ended boilers being 20.34 feet long and the single-ended boilers 11.98 feet long. Each double-ended boiler has six corrugated furnaces leading into three separate combustion chambers. There are seven hundred and fifty-six tubes, 3 inches outside diameter, making a total heating surface for the double-ended boilers of 5,200 square feet. Each single-ended boiler has three corrugated furnaces, leading to three separate combustion chambers, and three hundred and seventy-eight tubes, 3 inches outside diameter, making a total heating surface of 2,710 square feet for each single-ended boiler. Howden's forced draft is used, giving a pressure in the ash pits equivalent to 1.97 inches of water. The ship has a bunker capacity of 3,900 tons, with a reserve supply of 900 tons. Four ash ejectors supplied by Howald, of Kiel, are fitted in the boiler rooms.

Electricity for lighting and power throughout the ship is generated by seven separate steam-driven direct-connected generators, each having compound engines of 200 horsepower, with cylinder diameters of 12.60 and 22.05 inches and a stroke of 9.84 inches, operating at 220 revolutions per minute. The generators are each of 110 kilowatts capacity.



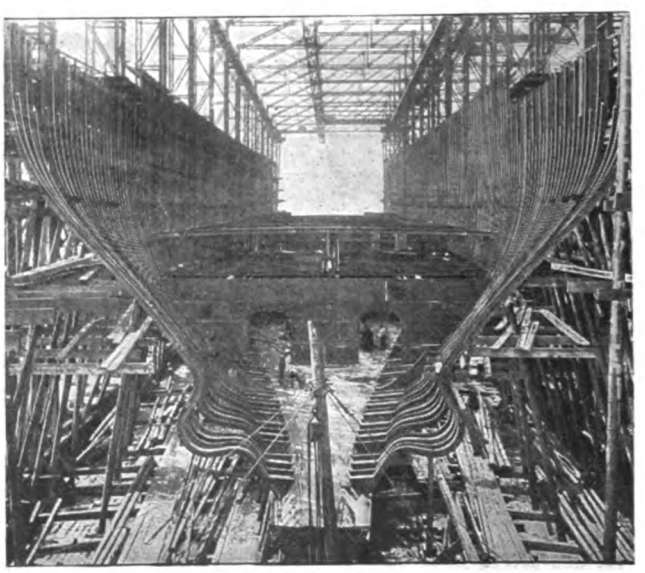
BULKHEAD CONSTRUCTION.

Ventilation is by thirty electrically-driven Sirocco fans.

The auxiliaries include six boiler-feed pumps, 17.32 by 12.60 inches, with a stroke of 26.57 inches and a capacity of 65 tons per hour. There are two steam-driven bilge pumps, each of which has a capacity of 102 tons per hour, the size of cylinders being 7.09 and 8.66 inches, and the stroke 13.78 inches. There are two other bilge pumps direct connected to the main engines, each of which has a diameter of 7.09 inches and a stroke of 21.65 inches. There are three centrifugal fire pumps, having a capacity of 8,480 cubic feet per hour, maintaining a pressure of from 118 to 147 pounds per square inch in the pipes. One hundred and fifty couplings for the attachment of hose are distributed throughout the vessel.

Two ballast pumps are provided, 8.66 by 11.81 by 13.78 inches, each having a capacity of 150 tons per hour; also sanitary pumps having a capacity of 190 tons per hour and three pumps for supplying water to the baths, one of which is steam driven, the other two being direct connected to the main engine. The steam-driven pump is 5.51 by 7.09 by 9.84 inches; and the direct-coupled pumps 5.91 by 21.65 inches.

Fresh water for the boilers is supplied by an evaporator having a capacity of 50 tons per twenty-four hours, while fresh drinking water is supplied by two evaporators, each having a capacity of 20 tons per twenty-four hours. The refrigerating plant includes one large machine and one auxiliary machine. The large machine is operated by a compound engine having cylinders 9.84 and 15.75 inches diameter, with a stroke of 11.81 inches. The small machine has a cylinder diameter of 5.91 inches and a stroke of 6.89 inches.



STERN FRAMING OF THE GEORGE WASHINGTON.

Cargo is handled through five cargo hatches, three forward and two aft, the deck machinery including 19 cargo winches, 1 capstan and 5 steam-driven windlasses. The cargo winches are driven by 7.89 by 13.78-inch engines. The windlasses are driven by simple engines having cylinder diameters of 13.78 inches and a stroke of 12.21 inches. The capstan is driven by a simple engine having a cylinder 18.50 inches in diameter and a stroke of 14.17 inches.

The steam steering gear consists of a Brown steam tiller and telemotor. The engine has two cylinders, each 12.99 inches in diameter, with a stroke of 11.81 inches.

The engine-room force on the ship consists of one chief engineer, one first engineer, two seconds, four thirds, four fourths and eight assistants. There are also four electricians,

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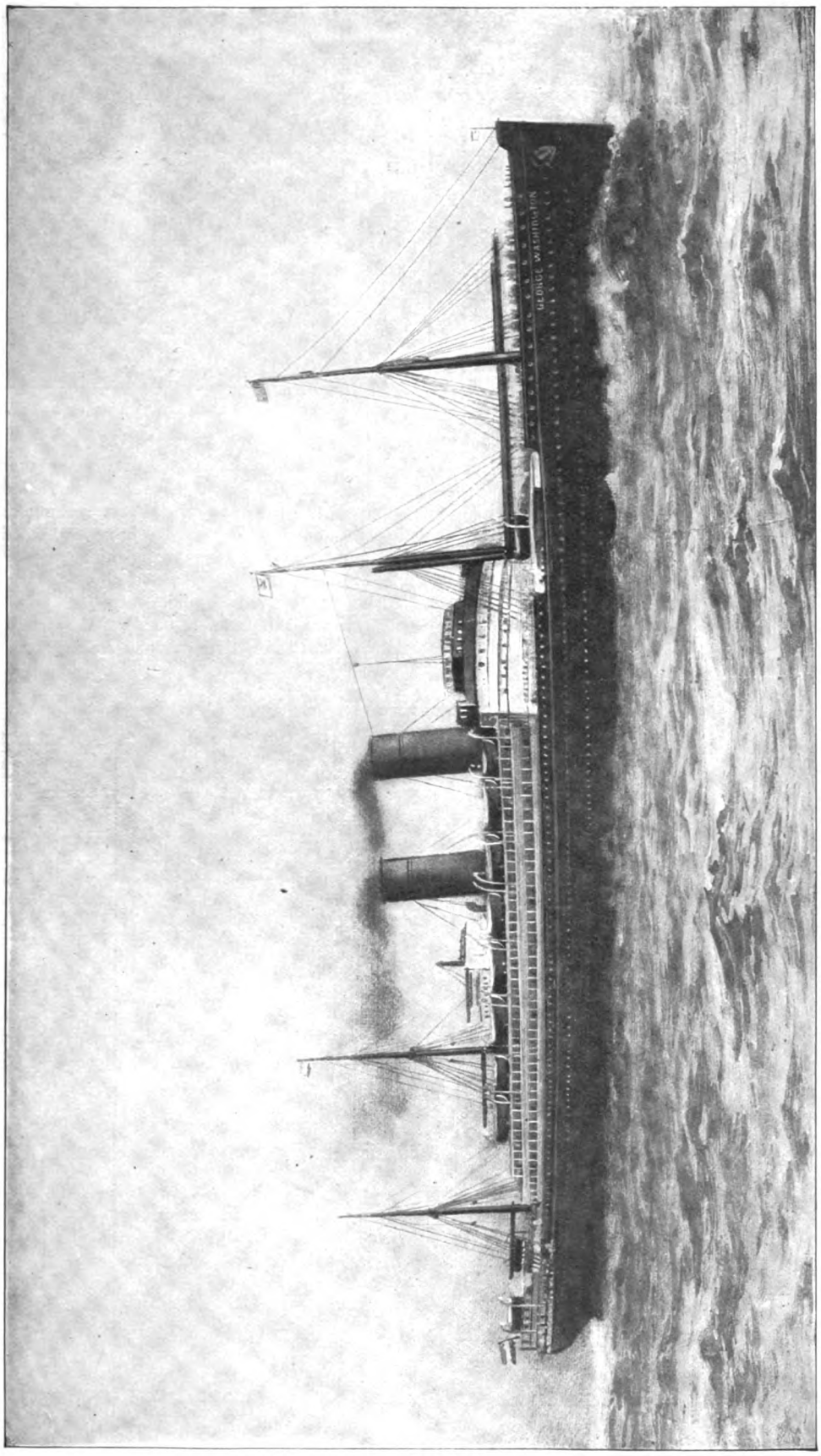

THE NEW NORTH GERMAN LLOYD LINER GEORGE WASHINGTON, THE LARGEST STEAMSHIP EVER BUILT IN GERMANY.

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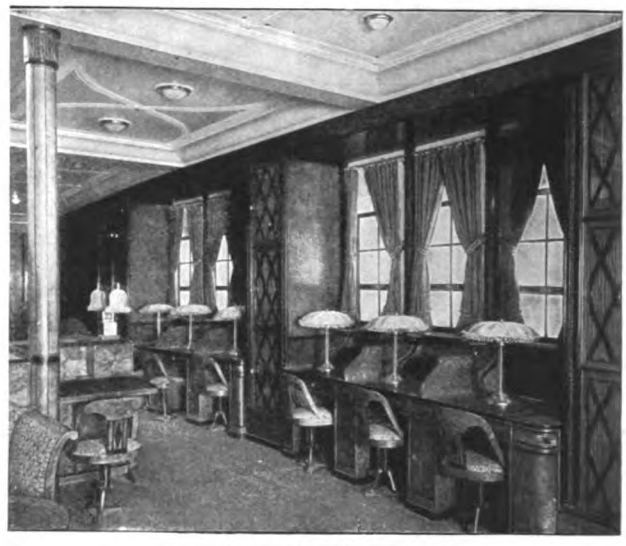
INTERIOR OF THE SMOKING ROOM.



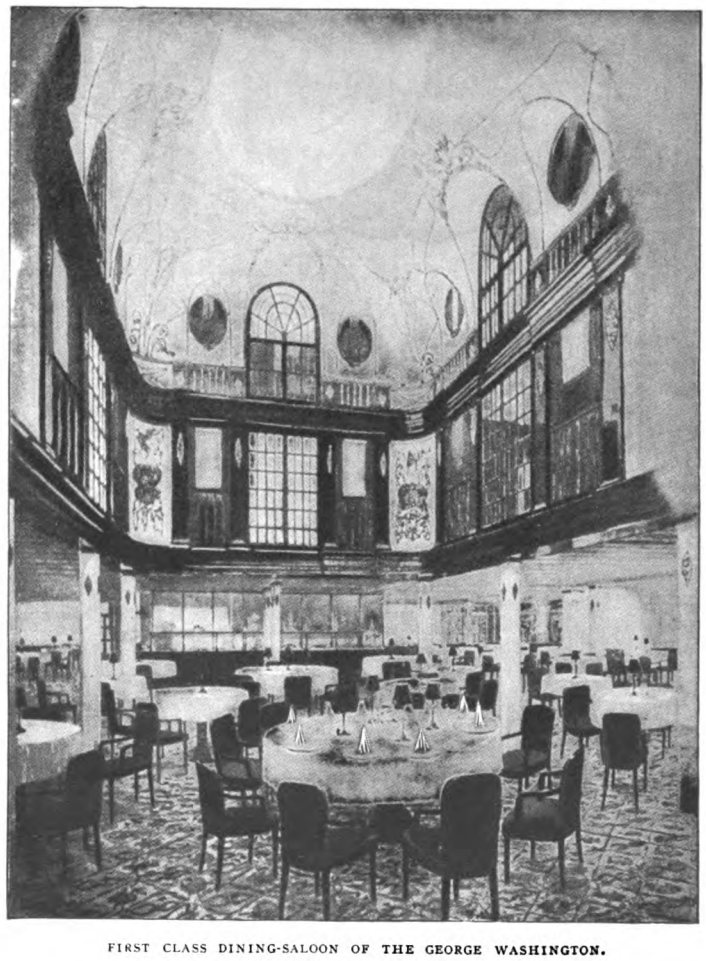
A CORNER OF THE WRITING ROOM.

six oilers, two storekeepers, one plumber and one boiler maker. The fire-room force includes six head stokers, sixty stokers and sixty-three trimmers.

Accommodation is provided on the vessel for first, second and third class and steerage passengers. The first class accommodations are all on the upper decks, amidships, with the second class aft on the upper and lower promenade decks. The steerage passengers are placed forward, while the third-class quarters are at the stern of the boat.

The arrangement and decoration of the first-class public rooms and staterooms have been carried out in a pleasing manner. The rooms are almost without exception of good height and tastefully paneled in natural wood. The first class dining room, in the center of which there is a well extending up through two decks, is especially attractive, the color scheme being red, white and gold.

The reading room, situated on the upper promenade deck, is decorated in quiet tones and writing tables and book cases



FIRST CLASS DINING-SALOON OF THE GEORGE WASHINGTON.

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are arranged around the walls and in the center of the room. The effect of great spaciousness has been obtained in this room by letting the book cases into the walls between the writing tables.

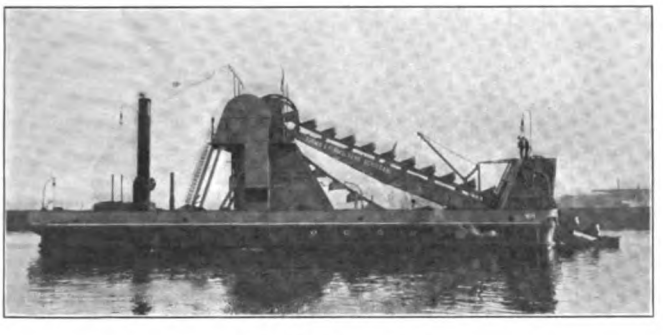
Numerous mural paintings, depicting the life and times of President Washington, have been supplied by Otto Bollhagen, of Bremen, who spent considerable time in the United States studying the scenes and history of the events portrayed.

Among the unusual features installed on the boat are two electric elevators, a large, well-appointed gymnasium and a nursery; also on the boat deck there are two specially-constructed dog kennels in charge of a competent kennel master.

Welin quadrant davits are used exclusively for handling the ship's boats.

BUCKET DREDGES No. ONE AND LA PLATA.

The bucket dredger *No. 1* is a type of stationary dredger built by the Werf Gusto, Shiedam, Holland, for the port of Copenhagen, while *La Plata* is a type of seagoing twin-screw bucket dredger, built by the same concern for Argentine. *No. 1* is 111 feet 6 inches long, with a beam of 26 feet 9 inches and a depth of 9 feet 9 inches, with a capacity of 275



BUCKET DREDGE NO. 1.

cubic yards of material an hour. She was especially designed for dredging clay and other rough material. Electric lighting is used throughout the vessel.

La Plata is 169 feet long, with a beam of 30 feet 6 inches and a depth of 12 feet 6 inches, with a capacity of 825 cubic yards of material an hour, dredging from a depth of 42 feet 7 inches.

The hull is divided into nine watertight compartments by means of eight bulkheads. The vessel is propelled by twin screws driven by two compound engines developing 260 indicated horsepower each, with surface condensation. Each of these engines may separately drive the bucket chains for dredging, and, while navigating, each engine drives one of the twin screws, giving the vessel a speed of about 7 knots. The auxiliary engines include five steam winches for the maneuvering chains fore and aft, for handling the anchors, for the hoisting of the bucket ladder and the chutes; an engine for driving the dynamo and a direct-working 6-inch centrifugal pump for supplying water to the chutes.

Steam is supplied by two boilers of the Scotch marine type, having a total heating surface of 2,600 square feet and a working pressure of 120 pounds per square inch.

The buckets each have a capacity of 25 cubic feet, so that when working at a speed of fifteen buckets a minute the dredger has a theoretical capacity of about 825 cubic yards per hour.

La Plata made the voyage from Europe to Buenos Ayres in forty days, and is to be used in deepening the harbors of Buenos Ayres and Rosario.

THE PROPULSION OF SHIPS BY MEANS OF CONTRARY TURNING SCREWS ON A COMMON AXIS.*

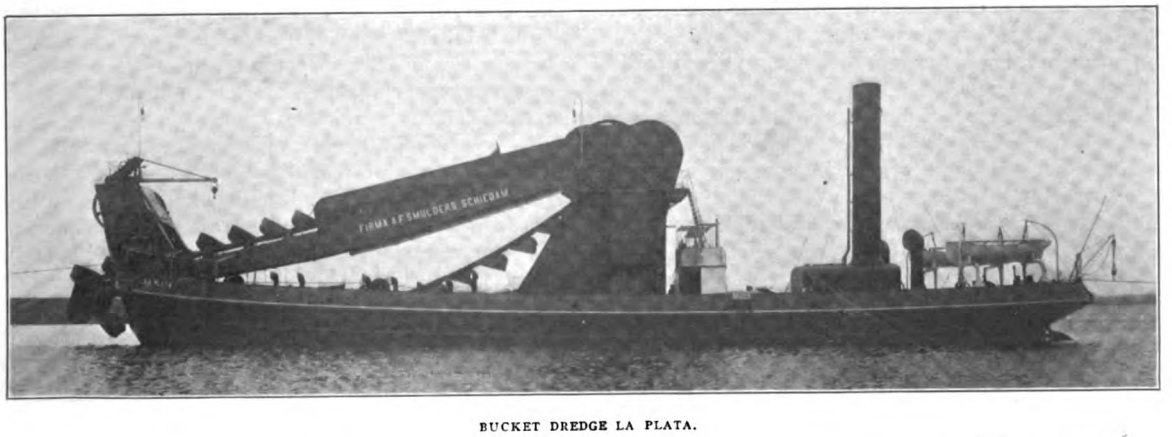
BY LIEUTENANT-COLONEL G. ROTA, R. I. N.

The method of propulsion by two contrary turning propellers on a common axis was first applied to marine purposes by Ericsson in the *Robert F. Stockton* in 1839. More recently we have had a general application of the same principle in the well-known Whitehead and similar torpedoes. In 1892 a little steamboat for passenger traffic on the Brent Lake (Neuchâtel) had a similar engine working the propellers by means of gear under the patent of Stengen, of Strasbourg. Another similar patent† was claimed for a double propeller on a common axis, in association with an oil engine and belt drive.

The guide blade of Thornycroft's system, and also the systems of Rigg and Parsons, can be considered as special cases of the double propeller arrangement, the after screw being replaced by guide blades. At the time that the above arrangements were devised, the reciprocating engine was the only kind of engine available for marine propulsion, and it was only possible to drive one of the two shafts (the inner one) off the engine, as no arrangement of gearing or belt driving was found to be practically workable for the outer propeller. The working screw was therefore placed in front of a fixed-blade device, which acted as a guide to divert the stream of water issuing from the screw in parallel lines astern.

But the fixed guide blades, independently of their beneficial influence on the efficiency, certainly increase the resistance of the hull. It is well known, however, that this increase of

* From a paper read before the Institution of Naval Architects, April, 1909.
† Described in INTERNATIONAL MARINE ENGINEERING, September, 1907.
‡ Trans. I. N. A., Vol. XXIX., p. 319.



BUCKET DREDGE LA PLATA.

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resistance can be reduced if the guide blades are not fixed, but form instead an integral part of the propelling apparatus. With such an arrangement the stream of water is conveyed to the after part of the propeller in a more convenient direction, thus ensuring the beneficial influence of that propeller on the forward one.

In his interesting paper of 1888,‡ Prof. Greenhill, in explaining his theory of the screw propeller, first analyzed the case of the double-screw propeller on a common axis, and pointed out the maximum efficiency it would be possible to obtain by a special arrangement for increasing the pitch of the screws. I venture to say that a gain of efficiency is generally obtained by the double-screw propellers on a common axis, not as a consequence of increasing the pitch under the Greenhill arrangement, but merely by the sub-division of the propeller into two parts acting in opposite directions, yet of constant pitch.

As the result of careful observations it may reasonably be assumed that an increase of efficiency is possible with the above arrangement of propellers instead of the ordinary single screw. It is well known that with an hydraulic turbine an essential condition to get good efficiency is to have the water conveyed by the guides before acting on the buckets; in that case the guides evidently act in the same way as the forward screw on the after one. Similarly, with steam turbines it is necessary, in every case, to have the acting fluid diverted from its ordinary course, and conveyed to the principal part of the moving apparatus in a direction more conducive to efficiency.

Turning to the question of the reciprocal influence of hull and propeller, a most convenient result is obtained with the double contrary turning screws on a common axis instead of the ordinary single screw.

(1) Because the increase of resistance caused by the action of the double propellers is lessened when compared with that obtained by the single screw, in consequence of the smaller diameter of the screws.

(2) Because the gain in wake is considerable in the case of the double screws.

Comparing the above arrangement of screws with ordinary twin screws, we also get:

(3) Reduced length of bracket for propeller shafts, and consequently great reduction in the resistance caused by brackets, which commonly produce a great additional resistance to the hull;

(4) Increase in the depth of water above the propellers, that causes another gain of efficiency.

Other beneficial effects will ensue when considering the question from the point of view of maritime and inland navigation.

Until now, although the arrangement of double contrary turning screws on a common axis is not new, its superiority as compared with the single screw has not yet been demonstrated in a practical manner. That has been the subject of my researches, and I am indebted to H. E., the Minister of Marine in Rome, for having permitted me to carry out a complete series of trials with a steamboat in the Royal Dockyard at Castellammare di Stabia, first with a single screw and afterwards with two contrary turning screws of different diameters on a common axis with a constant pitch, and also with increasing pitch according to Professor Greenhill's rules. The results obtained by these trials showed that a great gain of efficiency is to be expected with the double screws. It appears that about 20 percent of the horsepower can be saved.

In the experimental arrangement on the vessel which I had at my disposal for the trials, the two shafts are driven by gearing off the reciprocating engine, and the loss of power by friction should be taken into account in estimating the probable gain resulting from the use of the double screws. Considering the improvements that should be possible in designing a complete installation on the double-screw system, there is no doubt that a considerable gain in efficiency may be secured.

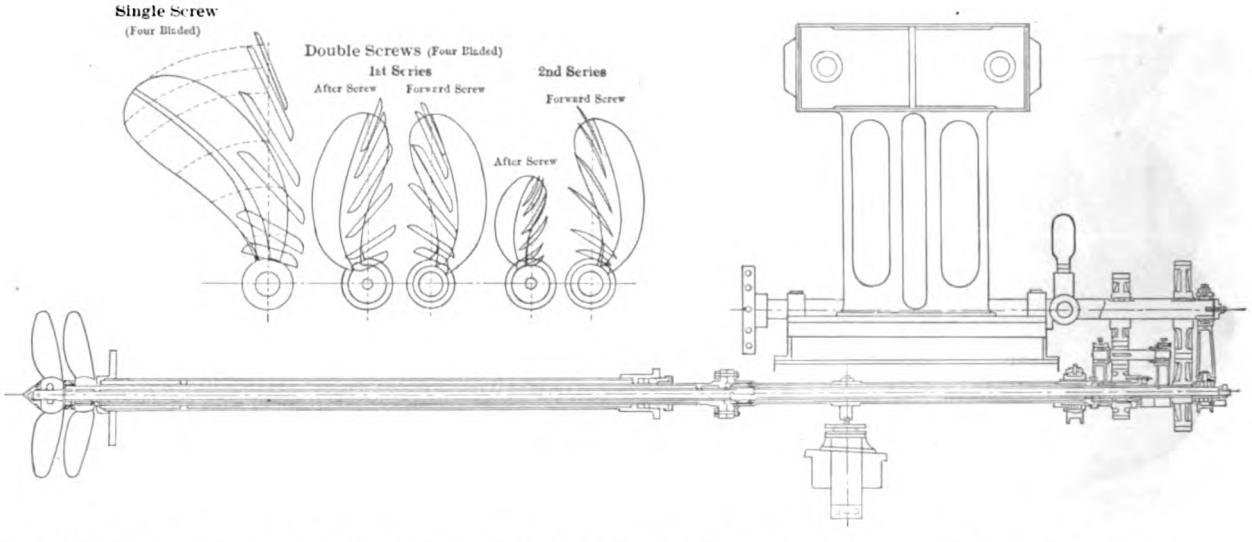

FIG. 1.—SHAPE AND DIMENSIONS OF THE SCREWS, AND GENERAL ARRANGEMENT OF ENGINE, SHAFTS, BEARINGS AND GEARS ON STEAMBOAT.

The main difficulty in the use of two propellers, one behind the other, and acting in contrary directions, may be now surmounted by the use of turbine engines, by which it would be possible to drive in opposite directions two shafts with a common axis, without gearing, belt, or other driving apparatus which is unsuitable for use on board ship. It might also be tried if the combination of steam turbines and reciprocating engines could be conveniently arranged to drive respectively the outer and the inner shafts, connected with the forward and the after screws respectively. The special turbine for reversing would thus be dispensed with, and the other advantages claimed for the "combination system" of machinery would be secured.

The special arrangement of propellers I have described is also of advantage in disposing of the conflict between steam turbines and screw efficiency. It is well known that steam turbines used on board ship cannot be so efficient as those

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used for land plants, it being impossible on board ship to have the high rate of revolution required for the highest possible efficiency. The screws have to be reduced in diameter, and consequently their efficiency is lowered, compared to that of the single screw driven by reciprocating engines. The special arrangement of propellers, the one behind the other, which I have described, allows of a greater number of revolutions, and consequently produces a higher efficiency of the steam turbines, and yet, with the reduced diameter of the screw, it allows a considerable increase over the propeller efficiency than is possible with a single screw. If, as I believe, the gain in power were not less than 20 percent in a cruising ship of moderate size, say, 10,000 tons, about 250 tons would

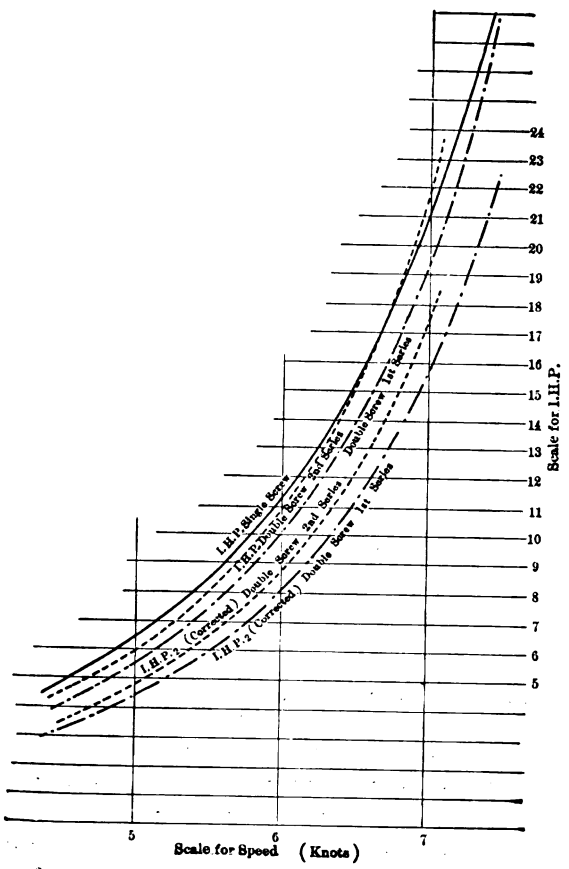
Scale for I.H.P.
Scale for Speed (Knots)
I.H.P. single Screw
I.H.P. Double Screw
I.H.P.1 (Corrected) Double Screw 2nd Series
I.H.P.2 (Corrected) Double Screw 1st Series
Double Screw 1st Series
Double Screw 2nd Series

FIG. 2.—COMPARATIVE RESULTS WITH SINGLE AND DOUBLE SCREWS.

be saved in weight of machinery required to obtain 22½ knots, or, on the other hand, a knot more speed could be obtained with the same horsepower.

The arrangement for a large installation of two shafts on a common axis and turning at high speed would, no doubt, involve difficult practical problems. The most convenient design of bearings for both shafts, the reliability of the apparatus, the maintenance of the different parts, the special structure of the hull in the vicinity of the shafts to prevent water entering in case of accident to the shafts, the lubricating arrangements, etc., should all be subjects of careful inquiry. The experimental arrangement of shafts and propellers which I installed on the steamboat for trials has, however, been working satisfactorily for a period of about a year in the ordinary service of the boat.

The boat on which the trials were carried out was of 25 tons displacement, 46 feet (14 meters) long, 11 feet 9 inches (3.6 meters) beam, equipped with a boiler having a total grate surface of 7.64 square feet (.71 square meter); heating surface, 224 square feet (20.8 square meters), supplying steam at a pressure of 75 pounds per square inch (5.3 kilograms per square centimeter) for an engine having cylinders 7.87 inches (.2 meter) in diameter with a stroke of 7.08 inches (.18 meter). The general arrangement of the engine, shafts and propellers is shown in Fig. 1.

Trials were first made with a single four-bladed screw 3 feet 9¼ inches (1.15 meters) in diameter, with a constant pitch of 4 feet 5 inches (1.35 meters). The double screw trials were run in two series with different propellers. In the first series the propellers were of similar shape to that used for the single screw, the combined blade area of the two screws equalling the blade area of the single screw and the number of revolutions being two times the corresponding number for the single screw. These screws were 2 feet 8 inches (.814 meter) in diameter, with a constant pitch of 3 feet 1½ inches (.954 meter).

In the second series of trials with double screws, the propellers were of the same general shape as the single screw, with a pitch according to Professor Greenhill's rules. The dimensions of the after screw were as follows:

	Feet.	Inches.	Meters.
Diameter	1	8¼	.516
Mean pitch	2	3	.69
Initial pitch	1	7	.485
Final pitch	3	10	1.17

The dimensions of the forward screw were:

	Feet.	Inches.	Meters.
Diameter	2	7	.791
Mean pitch	3	5¾	1.06
Initial pitch	2	11¼	.895
Final pitch	4	3	1.3

Fig. 2 shows the comparative results of the three series of trials and the corrections for extra loss of power for friction in the case of double screws as compared with single screws. The value of that extra loss of power for all speeds was estimated at 20 percent of the corresponding horsepower.

The advantage of the double contrary turning screws on a common axis is shown in the following table:

DOUBLE SCREW IN USE.

Speed of the Boat Knots	Single Screw in Use I.H.P.1	First Series. I.H.P.'	Extra Loss of Power for Friction Deduction.	I.H.P.2' (Corrected).	Gain of Power $\frac{\text{I.H.P.}_1\text{-I.H.P.}_2'}{\text{I.H.P.}_1} \cdot 100$ Percent.
5	6.3	5.45	1.09	4.36	30.5
5½	8.3	7.32	1.46	5.86	29.4
6	11.1	10.	2.	8.	28.
6½	15.15	13.7	2.74	10.96	27.6
7	21.	19.2	3.84	15.36	26.8

DOUBLE SCREW IN USE.

Speed of the Boat Knots	Single Screw in Use I.H.P.1	Second Series. I.H.P."	Extra Loss of Power for Friction Deduction.	I.H.P.2" (Corrected).	Gain of Power $\frac{\text{I.H.P.}_1\text{-I.H.P.}_2''}{\text{I.H.P.}_1} \cdot 100$ Percent.
5	6.3	6.	1.20	4.80	23.8
5½	8.3	7.90	1.58	6.32	23.8
6	11.1	10.80	2.16	8.64	22.1
6½	15.15	15.	3.	12.	20.8
7	21.	21.70	4.34	17.36	17.3

The Japanese Mercantile Marine.

In 1889 Japan had 3,536 merchant steamers, aggregating 768,538 tons. In 1908 the number of steamers had increased to 6,098, and the tonnage to 1,494,676. The increase in Japan's mercantile marine during the past ten years has been as follows: 1899, 3,536 steamers, 768,538 tons; 1901, 4,534 steamers, 902,190 tons; 1903, 4,624 steamers, 977,308 tons; 1905, 5,089 steamers, 1,260,087 tons; 1907, 5,784 steamers, 1,462,718 tons; 1908, 6,098 steamers, 1,494,676 tons.

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What Would You Do?

As chief engineer of a steamship, what would you do if the low-pressure cylinder head cracked while the vessel was under way? If a furnace collapsed, or a tube burst, what would you do? What would you do with a vessel whose boilers primed excessively, or with one where corrosion was particularly active in the boilers in spite of every precaution that could be taken? What would you do if the tail shaft broke, or if the propeller dropped off, necessitating the fitting of a spare shaft and propeller? How would you repair a cracked tube sheet while at sea? In the event of the condenser or circulating pumps failing, how would you rig up a jet condenser? Could you build a makeshift pump piston to replace a broken one? What would you do if a fracture developed in the main steam pipe or in any of its valves and fittings?

Questions such as these are not mere matters for idle speculation while coaling ship or lying-to in a safe harbor, but they are questions which frequently call for immediate answer under the most trying circumstances. Breakdowns are bound to occur even on ships where the greatest care is used in the maintenance of the machinery, and it is absolutely necessary that the chief engineer and his assistants should be men competent to face and solve such problems in a satisfactory way. Aside from the fact that it takes a good mechanic to plan and execute a satisfactory repair job on any piece of machinery, the quality most to be desired in a marine engineer, who is likely to be called upon to face such emergencies, is resourcefulness.

Outside of a natural aptitude for mechanics and a thorough training in steam engineering and allied subjects, the thing which makes one engineer more resourceful than another is the fact that he has a wider knowledge of what has been done by other men in similar circumstances on other ships. Stories of breakdowns and how they were repaired are seldom brought to the public notice, on account of the natural reticence of shipowners and marine engineers regarding accidents which have happened to their ships, and consequently the opportunities for most engineers to obtain such information are somewhat limited. During the past few months we have had an opportunity to secure complete details covering a considerable number of repair jobs, several of which are described in this issue. These will be followed by others of the same general character in subsequent issues, and we hope that these few instances will serve to bring forth many more similar articles from our readers.

Superheated Steam.

Superheated steam is no novelty. For over half a century it has been used to a greater or less extent in steam-power plants of all descriptions. During this time it has been the subject of extended and thorough investigation not only as to its individual properties and behavior under various conditions, but more particularly in connection with its use as a factor of economy in steam-power plant design. The whole subject has been so thoroughly discussed time and time again that it would be superfluous to again refer to it if it were not for the fact that current progress in the development of steam boilers and engines is continually bringing forth new conditions and questions in relation to the economy of steam-power plants in which the use of superheated steam is a factor which cannot be disregarded. For one thing, the introduction of the steam turbine has emphasized more than ever before the advantages to be gained by the use of superheated steam, as it has been shown that the reduction in steam consumption of turbines due to the use of superheated steam is very materially greater than the reduction in the steam consumption of reciprocating engines due to the same cause, at least in land installations, and there is no reason why the same should not hold true in marine work. According to data collected by Mr. R. M. Neilson, which is quoted elsewhere in this issue by Mr. F. J. Rowan in his resumé of superheated steam as applied to marine work, the increased economy of the turbine over the reciprocat-

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ing engine due to the use of superheated steam is apparent at both low and high degrees of superheat, the percentage reduction of steam consumption per degree Fahrenheit ranging all the way from 0.47 at 13 degrees to 0.09 at 260 degrees in the case of the turbine, and from 0.25 at 31 degrees to 0.07 at 440 degrees in the case of the reciprocating engine. In general, it is claimed that the steam consumption of the turbine may be reduced 1 percent for each 10 degrees F. of superheat. Designers cannot well afford to disregard claims such as this when considering the installation of turbines as prime movers.

Many years ago Rankin summed up the advantages of using superheated steam as follows: an increase in the efficiency of the motive fluid without producing a dangerous pressure; a diminution of the density of the steam and consequent lessening of the back pressure, and a reduction of cylinder condensation and leakage. It is now pretty generally understood that practically the whole benefit derived from superheating is due to the latter cause. The thermodynamic advantage is comparatively small. The real advantage gained by superheating can only be determined by taking into consideration the performance of the entire plant, including the boiler, superheater, steam piping, valves and engine. An added consideration involved in marine installations is the question of weight and space occupied by the superheating apparatus. This is a matter of little importance on shore, but on board ship it cannot be neglected. Closely associated with this is the question of the effect of superheated steam on the design of the engine, piping and valves, and also on the liability of breakdowns and the increased cost of repairs to the machinery and the cost of the maintenance of the superheater itself. As regards the real advantage to be gained by the use of superheated steam in marine work, most results which have been published, where there is an opportunity to compare the performance of the same plant with and without the superheater, seem to show that an average saving of about 15 percent in coal consumption can be obtained by superheating. This is, of course, a very general statement, as the results in any particular case may vary widely from this, depending upon local conditions. This gain in coal consumption is apparently not obtained at the expense of any great increase in the cost of repairs or maintenance of machinery, for progress in the improvement of the materials and of the design of engines, valves, piping, etc., and in the construction of the superheater itself, has kept pace fully with the increased demand for apparatus to successfully withstand the high temperatures involved by the use of superheated steam. In the earlier days of superheated steam troubles with superheater tubes and troubles with rubbing surfaces of cylinders, piston rods, valves and packings frequently led engineers to abandon superheated steam for the many less troublesome means at their disposal for improving the capacity and economy of their steam plants. Now, however, since most of the refinements which were then available in other directions have become matters of standard practice, it is impossible to ignore the advantages of superheat, as far as the economy of a marine steam plant is concerned. That superheaters are now being considered as a necessity in a great many cases is shown by the fact that in America alone no less than twenty vessels, aggregating 158,450 horsepower, are so equipped, eight of which are naval vessels.

Progress has been made not only in the design of superheating apparatus and in the adaptation of engines to the use of superheated steam, but also in the investigation of the properties of superheated steam. For a great many years Regnault's figure of 0.48 was relied upon as the correct mean value of the specific heat of the steam; but recent investigations carried out in Germany by Knoblauch and Jacob and in America by Drs. Thomas, Heck and others, have shown that there is a considerable variation in the specific heat with different conditions of pressure and temperature. In general, the value of the specific heat decreases for any pressure as the temperature rises and increases for any given temperature as the pressure rises. Results obtained by Dr. Thomas, in America, and by Mollier, in Germany, agree substantially with those obtained by Knoblauch and Jacob. Furthermore, recent experiments by Dr. Harvey N. Davis, of Cambridge, Mass., have shown that Regnault's classic formula, which for sixty-one years has been used for the total heat of saturated steam, is not correct. Recomputed values of the specific heat of saturated steam differ from former standards by 3 percent at 32 degrees and by about 1 percent in the opposite direction at 275 degrees. Below 212 degrees there is an abundance of modern data to show that Regnault's formula runs high, the error reaching 18 B. T. U. at 32 degrees. Calculations based on the old values as established by Regnault are not entirely worthless, but now that more accurate values are at hand all computations of importance should be based on the new values.

At the present time there are not many different types of superheaters on the market, seven or eight different designs covering the entire field. It has been generally understood that the use of watertube boilers gave the superheater a decided impetus, because the superheater could be more easily applied to that type of boiler. Such is not really the case, however, for most of the superheaters can be applied to cylindrical boilers of the Scotch type. In general, it has been found an advantage to have the superheater so constructed that it could be cut off from the path of the hot gases if desired. Independently-fired superheaters have been used on board ship to some extent, although their use is more recent than the other types, the earliest types of superheaters usually being fitted in the uptakes or at the base of the funnels.

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Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Tons.	Knots.		May 1.	June 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons	90.0	92.3
Michigan ...	16,000	18½	New York Shipbuilding Co..	97.4	98.1
Delaware ...	20,000	21	Newp't News Shipbuilding Co.	77.9	82.4
North Dakota	20,000	21	Fore River Shipbuilding Co..	81.5	84.8
Florida	20,000	20¾	Navy Yard, New York	11.9	16.4
Utah	20,000	20¾	New York Shipbuilding Co...	14.9	20.0
TORPEDO-BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons	81.6	88.4
Lamson	700	28	Wm. Cramp & Sons	75.7	80.5
Preston	700	28	New York Shipbuilding Co..	70.7	77.4
Flusser	700	28	Bath Iron Works	68.7	74.0
Reid	700	28	Bath Iron Works	67.8	73.0
Paulding ...	742	29½	Bath Iron Works	9.8	14.2
Drayton	742	29½	Bath Iron Works	9.7	14.2
Roe	742	29½	Newp't News Shipbuilding Co.	38.6	46.7
Terry	742	29½	Newp't News Shipbuilding Co.	33.7	41.0
Perkins	742	29½	Fore River Shipbuilding Co..	22.0	28.3
Sterrett	742	29½	Fore River Shipbuilding Co..	22.0	28.3
McCall	742	29½	New York Shipbuilding Co..	11.7	13.1
Burrows	742	29½	New York Shipbuilding Co..	11.3	12.8
Warrington..	742	29½	Wm. Cramp & Sons	16.0	19.6
Mayrant	742	29½	Wm. Cramp & Sons	16.1	23.4
SUBMARINE TORPEDO BOATS.					
Stingray	...	..	Fore River Shipbuilding Co..	89.8	91.7
Tarpon	...	..	Fore River Shipbuilding Co..	89.7	91.7
Bonita	...	..	Fore River Shipbuilding Co..	81.4	85.2
Snapper	...	..	Fore River Shipbuilding Co..	80.4	84.9
Narwhal ...	...	..	Fore River Shipbuilding Co..	89.7	91.6
Grayling ...	...	..	Fore River Shipbuilding Co..	84.6	88.8
Salmon	...	..	Fore River Shipbuilding Co..	75.3	81.0
Seal	...	..	Newp't News Shipbuilding Co.	12.7	18.0

TECHNICAL PUBLICATIONS.

Directory of Shipowners, Shipbuilders and Marine Engineers, 1909. Size, 6 by 8½ inches. Pages, 749. London: The Directory Publishing Company, Ltd., 3 Ludgate Circus Building, E. C. Price, 10/ ($2.50).

This important directory has reached its seventh year of publication and fully justifies its existence; in fact, it seems indispensable to all who would be fully acquainted with the personnel of the shipowning, building and marine engineering world. All the shipping companies are detailed, with particulars as to their boats; to facilitate reference there is a copious index, and the owner and details of any particular vessel can be easily found. The leading officials of the various firms of shipbuilders and marine engineers are given, together with the number of berths, maximum output, capacity, and the size of the drydocks. Besides the "Boat Index," already indicated, there is a "Personal Index."

Machine Drawing and Design for Beginners. By Henry J. Spooner, C. E. Size, 9 by 6¾ inches. Pages, 266. Illustrations, 751. London, 1908: Longmans, Green & Company. Price, $1.25.

The author of this book has had extended experience in teaching the subjects of mechanical drawing and machine design, and has endeavored to present the subject in this book in such a way that the student may learn the rudiments of the subject without the aid of outside instruction. For this purpose the first six chapters are devoted entirely to the subject of drawing, the simplest details being carefully described and illustrated. It is intended that these chapters shall give the student sufficient instruction to enable him to learn the art of making working drawings of simple pieces. The remaining chapters treat more particularly of matters relating to details of machine parts. Not only is the correct method of making working drawings of the details explained, but the calculations for proportioning the various parts are also given, so that it is not necessary for the student to refer to a handbook or other source of information for his data. The details described are very practical, including such pieces of machinery as shafting, cranks, journals, couplings, clutches, keys, bolts, nuts, screws, roller and ball bearings, toothed gearing, pistons and cylinders, crossheads and guides, connecting rods, etc. One chapter is devoted to riveted joints, in which not only the proportions of different kinds of rivets are given, but the computations for various styles of riveted joints, and the method of figuring strength is also described. The text is concise and clear, and the book is profusely illustrated, so that it cannot fail to be of value to the student in this branch of engineering.

Steam Turbines. (Power Handbook Series), by Hubert E. Collins. Size, 4½ by 6¾ inches. Pages, 186. Figures, 76. New York, 1909. Hill Publishing Company. Price $1.00.

The large number of steam turbines now being installed in power plants necessitates the acquisition of a certain amount of practical knowledge regarding turbines by the engineer in charge of the plant in order that he may operate the plant successfully. This book attempts to give a compact manual for engineers who are in charge of turbine plants. The principal standard types of turbines are carefully described and following this are chapters on the proper methods of testing a steam turbine, auxiliaries for steam turbines and troubles with steam turbine auxiliaries.

SIR WILLIAM HENRY WHITE.

Probably there is no more eminent living authority on the subjects of naval architecture and marine engineering than Sir William Henry White, K. C. B., F. R. S., LL. D., D. Sc., formerly chief constructor of the British Admiralty. He first became connected with the Admiralty in 1867, and remained in the constructive department until 1883, rising meanwhile to the rank of chief constructor. From 1870 to 1881 he was also professor of naval architecture at the Royal School of Architecture and Royal Naval College. When, in 1883, Armstrong & Company, of Newcastle, established a department for the building of warships, Sir William White organized and directed the department. This work occupied him until 1885, when he became director of naval construction and the assistant controller of the Royal Navy. This position he held until February, 1902, and, during that period, he was responsible for the design of all His Majesty's warships. He was finally forced to resign on account of ill health, and was awarded a special grant of money by vote of Parliament in recognition of his exceptional services to the navy.

Such, in brief, is the professional record of this distinguished constructor. That his time has not been devoted entirely to the pursuit of one single branch of science is evident from the interest which he took in almost every important engineering organization, not only in Great Britain, but, as well, in the United States and other countries. He is a past president of the Institution of Civil Engineers and of the Institution of Mechanical Engineers, a vice-president of the Institution of Naval Architects and past president of the Institutions of Marine Engineers and Junior Engineers. He is also past master of the Shipwrights Company of London; a foreign member of the Royal Academy of Sciences of Sweden; and an honorary member of the Association Technique Maritime, the American Societies of Civil Engineers, Mechanical Engineers and Naval Architects; and an honorary member of the Institution of Engineers and Shipbuilders in Scotland; of the Northeast Coast Institution of Shipbuilders and Engineers, and chairman of the Board of Studies in mechanical engineering at the London University.

Numerous professional papers have been presented by him before these various societies, and he has contributed much in the way of valuable discussion on subjects presented by other members. He is the author of a "Manual of Naval Architecture," and also a frequent contributor to the technical and engineering press.

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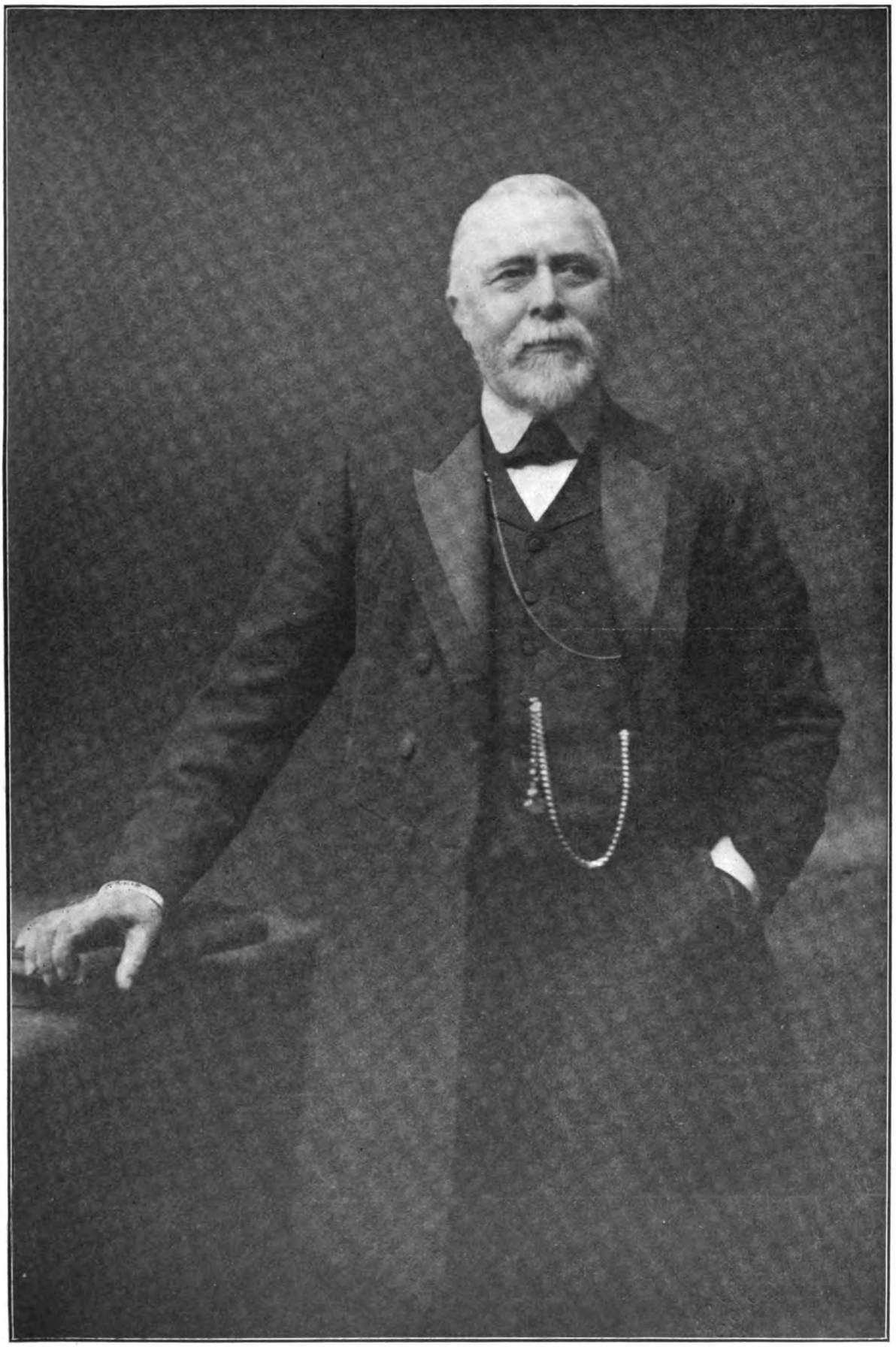

SIR WILLIAM HENRY WHITE, K. C. B., F. R. S., LL. D., D. SC.

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ENGINEERING SPECIALTIES.

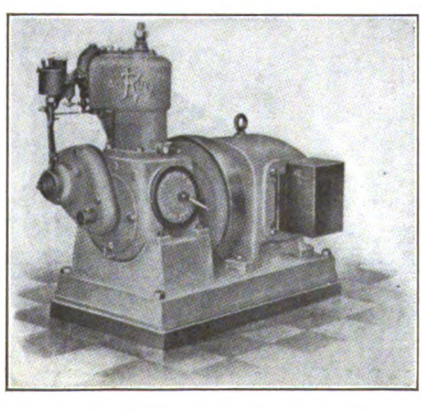
Standard Piston Rings.

The piston rings manufactured by the Standard Piston Ring & Engineering Company, Limited, Premier Works, Don Road, Sheffield, are so constructed that the spring combines the necessary vertical and lateral actions in such a way that the rings can be worked at the highest speeds and pressures commonly used. This is due particularly to the fact that a maximum amount of vertical pressure is obtained against the piston flanges. The inner surface of the piston ring is coned to afford a snug bed for the oval springs when in compression. As shown in the illustration, these springs lie immediately under the face of the ring in contact with the cylinder. They

have liberal bearing surface, so they do not form grooves in the piston rings, yet they may be accurately adjusted by a thin washer placed over the shanks of the long springs. The packing rings themselves are made from a special brand of iron manufactured exclusively for piston rings. It is claimed that it is tough and very close-grained, and possesses good enduring qualities. Tests are quoted by the manufacturers, showing that one of these rings of the Ramsbottom type, 3¼ inches in diameter, and having an opening of ⅛ inch, was sprung open until the opening was 1⅛ inches, the extension being, therefore, 1 inch. This was again tried, the width of the opening being extended ⅛ inch at a time until it reached 1½ inches.

The Reavell Vertical Paraffin (Kerosene) Oil Engine.

The Reavell paraffin oil engine is of the vertical "four stroke" type, having one power stroke in every two revolutions. The cylinder of the engine is directly bolted to the standard, and the main bearings are spigotted into an accurately bored recess on either side, thus ensuring good alinement. The half-time shaft works in bushes also fixed in the standard, the cams being made solid with the shaft. Both inlet and exhaust valves are mechanically operated from this shaft and are of ample area. A special arrangement is provided to prevent any but a vertical thrust being exerted on the tappet rods. This consists of levers carrying rollers, having their fulcrum at the side opposite to the cam shaft, thus obviating undue wear on the bushes which guide the tappet rods. The governor is of the horizontal centrifugal type directly attached to the shaft and is arranged on the end of the half-time shaft, being connected by means of suitable levers to a butterfly valve fixed in the vapor-inlet pipe. Governing is effected by throttling the supply of vapor. The commutator is of standard design, consisting of a spring-loaded roller pressing over a metal segment. The position of the commutator can of course be set so as to give the most efficient results for igniting the charge. The engine is water-cooled and can be arranged either with a thermo-syphon tank, circulating pump, or any other convenient method. The manufacturers are Reavell & Company, Limited, Ipswich.



A Few New Starrett Tools.

The illustrations show a number of new tools recently placed on the market by the L. S. Starrett Company, Athol, Mass. Fig. 1 is a universal bevel protractor constructed with verniers reading to five minutes or one-twelfth of a degree. The verniers are so placed with relation to the graduated half-circle as to make the protractor readable by vernier in any

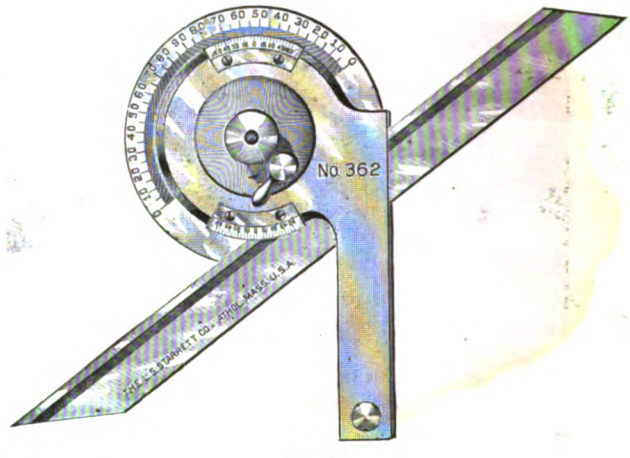


FIG. 1.

position. The disk is graded in degrees from zero to 90 each way and rotates the entire circle on a central stud inside the case. The blade clamped by an eccentric stud against the edge of the disk may be slipped back and forth its full length, or turned at any angle around the circle and firmly clamped at any point. An important feature of the tool is the fact that

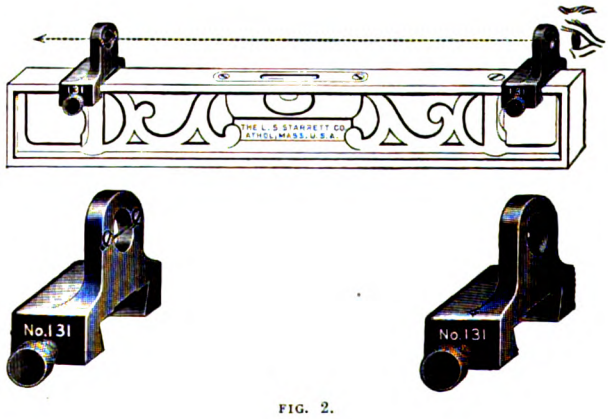


FIG. 2.

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the figures on the vernier are placed close to the holes, thus making it easy to read the tool when taking measurements. The central lock nut may be given a slight turn when the protractor is firmly held in position. The protractor stock is 4 inches long and has either a 7 or a 12-inch blade ½ inch wide; with the 7-inch blade the tool weighs only 6 ounces.

Fig. 2 shows attachments made to slip on and off the top side of the Starrett Company's iron levels to be used for sighting. The attachments are held in place by set screws and are provided with sight holes, one with a cross wire, en-

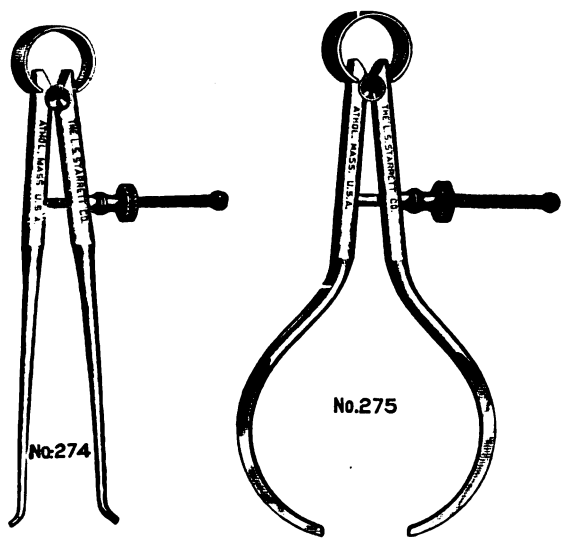


FIG. 3.

abling the workman to line accurately from the top of and parallel with the level. Sighting through the holes will enable one to use the common level for leveling a plot of ground from a fixed point at long range. These sights are made in sizes corresponding with various sizes of levels.

Fig. 3 shows two styles of tool makers' calipers. These are made from round stock with the legs drawn down, making them stiff and hard. The fulcrum stud is hardened and the screw and nut are carefully fitted. These tools are made in sizes from 2 to 6 inches.

Robinson's Rotary Cutters and Suction Apparatus.

Editor INTERNATIONAL MARINE ENGINEERING:

I am interested to note that three of the dredges illustrated in your May number are from my designs, namely: The 8-yard dipper dredge for the Cuban Government, the hydraulic dredge *Alexandra*, built by Simons & Company, and the two large dredges *Jinga* and *Kalu*, for Bombay, also built by Simons. These three dredges built by Simons are fitted with my improved dredging apparatus, including rotary cutter, suction frame and driving gear, for which the details were furnished from this office, and the remainder of the ship details, etc., being furnished by Simons. In your article describing the *Alexandra* I observe you give me credit for Robinson's rotary cutter, but in the article describing the *Jinga* and *Kalu* there is no credit given. I may say that since your article was written these two dredges for Bombay have completed their three months' guarantee, and having fulfilled the requirements have been accepted by the Bombay Trust. The work done at times exceeded 3,000 yards per hour pumped a distance of 4,000 feet.

MONTREAL, CAN. A. W. ROBINSON.

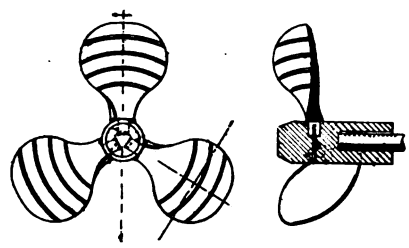
SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

914,857. PROPELLER. MORGAN B. MILLER, OF SAN JOSE, CAL., ASSIGNOR OF ONE-HALF TO GEORGE W. HARVEY, OF SAN JOSE.

Claim 1.—A propeller blade having a dished engaging face, the curvature thereof being on arcs of circles and having a greater radius



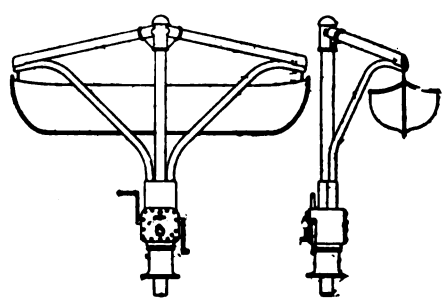
in a direction transverse from the propeller shaft than radially thereto. Four claims.

915,004. PNEUMATIC PROPULSION OF VESSELS. EDWARD WILDE, OF PHILADELPHIA, PA.

Claim 1.—In a means for the propulsion of vessels by pneumatic pressure, the combination of a vessel having a keel, of a guard at each side of the keel, and for a portion of its length, and below the hull; spaces formed between the guards and the keel which are open to the circulation of water at the bottom and at each end, and compressors having means to deliver pneumatic pressure to said spaces, toward the bow and toward the stern of the vessel. Eight claims.

915,118. BOAT DAVIT. PERCY G. SANBORN, OF SAN FRANCISCO, AND WALTER A. HESSE, OF ALAMEDA, CAL.

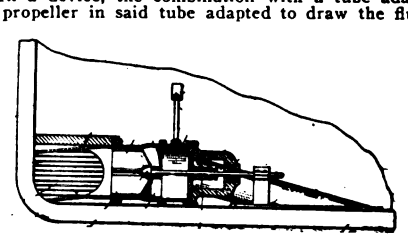
Claim 1.—In combination with a stand or base, a casing rotatable on said base, means for rotating the same thereon, a post secured upon said casing and having diverging arms arranged to guide ropes for



supporting a boat, drums in said casing on which said ropes can be wound, high and low-speed mechanisms for rotating said drums, and means for selectively operating said mechanisms. Eleven claims.

915,255. SCREW PROPELLER. ALBERT RICHARD WEISZ, OF BROOKLYN, N. Y., ASSIGNOR, BY DIRECT AND MESNE ASSIGNMENTS, TO WEISZ ROTARY PISTON AND ATMOSPHERIC MOTOR COMPANY, A CORPORATION OF NEW YORK.

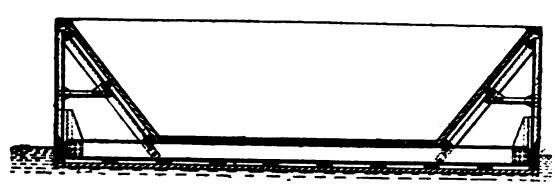
Claim 1.—In a device, the combination with a tube adapted to conduct fluid, a propeller in said tube adapted to draw the fluid into said



tube at one end and discharge same at the other end; of means disposed in the suction side of said tube conforming snugly to the shape of said propeller for preventing a whirling motion of the fluid before it reaches said propeller. Two claims.

915,410. BARGE. ARTHUR M. BOWMAN, OF AVALON, PA.

Claim 2.—In a barge, the combination of a wooden bottom having partly submerged sides, bottom planking, inner transverse metallic beams, metallic sides projecting above the wooden sides, inwardly ex-



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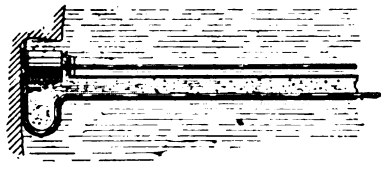
tending side-bracing elements connected with the metallic sides, upwardly rounded end plates, and intervening transverse partitions. Eight claims.

915,454. DREDGER HEAD. ROBERT A. LOWE, OF DULUTH, MINNESOTA, ASSIGNOR OF ONE-HALF TO W. H. LAMSON, OF HINCKLEY, MINN.

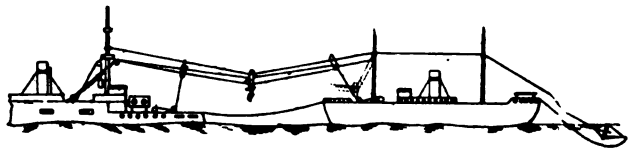
Claim 2.—A dredger head having an intake opening and a nozzle located on the wall of said head and arranged to project a jet against



the material contiguous to said opening and a material carrier arranged to deliver the material through said opening. Eighteen claims.

915,458. POWER TRANSMITTER AND CONTROLLER. THOMAS SPENCER MILLER, OF SOUTH ORANGE, N. J.

Claim 1.—In combination, a motor, two actuators by which its power is delivered, a driving connection between the motor and the first one



of said actuaries, a transmitter between the first and second of said actuators, a movable transmitter carrier and means whereby a uniform strain may be applied to said transmitter. Fifty-one claims.

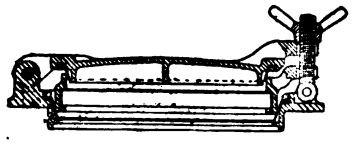
British patents compiled by G. F. Redfern & Company, chartered patent agents and engineers, 4 South street, Finsbury, E. C., and 21 Southampton building, W. C., London.

23,024. SHIPS' BULKHEADS. W. M. HOSKINS, BORDESLEY, BIRMINGHAM.

Portable partitions, for use in ships for the purpose of dividing a space into cabins, are of such a length that they fit snugly between the stanchions. The edges are shaped to fit closely to the stanchions, and the rear sides are left open to permit the bulkheads to be placed in position. Open-topped cup brackets on a casting are adapted to receive the snugs on the fitting secured to the bulkhead. In order to secure privacy, a flange or closure plate is provided on the fitting, or attached to the bulkhead, completely covering one side of the bulkhead clearances. The cup brackets are provided with extensions or bearing pieces, against which the outer vertical edges of the closure plates are adapted to fit. The bulkhead fittings may also be provided with sockets to receive studs carried by folding berths, by which the berths may be supported.

23,244. SHIPS' SCUTTLES AND SIDE LIGHTS. F. G. P. PRESTON, OF STONE & CO., DEPTFORD, KENT, AND G. E. JAKEMAN, PECKHAM, SURREY.

A jointing device, for the scuttles and side lights of ships, consists of a ring of metal or other material having a certain amount of elasticity, comparatively thin, and mounted in the frame of the scuttle, so that the edge of the glass frame, which is made sharp, beaded, or of conical form, may contact with the surface of the ring and form a tight joint when pressed against it. The figure shows a form of the jointing ring of the shape of a frustrum, applied to both the glass frame and dead light. Instead of this form, the ring may be of either of several different forms; in one of which are two contacting rings arranged to



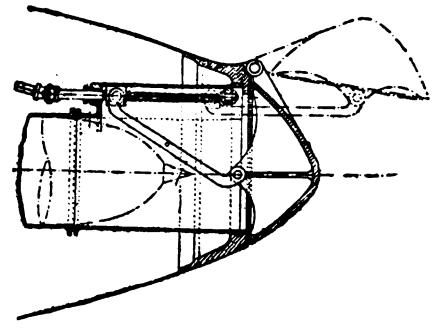
form a tight joint under pressure applied when closing the scuttle. Another form of the ring is of semi-circular cross-section, and is reinforced by a backing of rubber or other resilient material. When applied to frames which turn on trunnions through a right angle to admit air through the scuttle, the ring is bulged or curved in section. The jointing device is also described with reference to sliding scuttles and to scuttles closed by partial rotation in conjunction with wedge action.

23,530. LIFEBOATS. G. E. ENGLUND, LYSEKIL, SWEDEN.

In lifeboats of the class in which the boat proper is supported by an hermetically-closed spool-shaped floating body, the spool-shaped body is provided with a contracted central part, around which is rotatably mounted a drum, rigidly connected to the boat proper, and keel. To prevent excessive friction between the drum and the central part, anti-friction rings are provided, which are prevented from longitudinal movement by means of stop rings. Rollers or balls may be used instead of the rings. The boat may be provided with watertight compartments in its bottom and with oars or other propelling apparatus. The boat is independent of the rotary motion of the float, and capsizing is thus prevented.

23,518. TORPEDOES. P. M. JUSTICE, LONDON.

Ejecting Tubes.—The door is closed by means of a link and a screwed spindle and nut operated by a rod. A screwed nut on the rod is adapted to lock the firing lever when the door is closed. The lock bolt



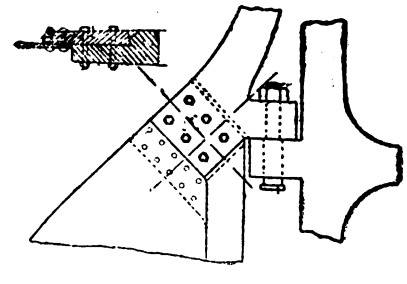
and the firing valve are operated by a common lever. A slot-and-pin connection is provided, so that the bolt may be removed before the valve is opened.

23,742. SHIPS' MASTS AND FUNNELS, LOWERING AND RAISING. H. CORDEN, NEW HOLLAND, LINCOLNSHIRE.

The invention is described as applied to ships' funnels, and is stated to be applicable to masts. The upper part of the funnel is pivoted in a spindle supported by the lower part of the funnel. The funnel is lowered by means of a wormwheel or toothed quadrant fixed on the spindle or on the disc attached to the spindle, and engaging with a worm on a spindle turned by means of a hand wheel. The spindle is mounted on bearings in a hollow casting or box carried by a bracket on the base of the funnel. The box may be pivoted on the bracket, and the bracket may have a radial slot in it for studs on the box to work in, so that the box and the spindle may be adjusted and set at any suitable angle. Means are employed for locking the funnel to the base part when the funnel is in its raised position, so as to take the weight of the funnel off the spindle. A link having side projections is carried by a bracket mounted on the funnel, and a bracket mounted on the bracket has two lugs, the faces of which are inclined, so that the bottom part of the link will ride up and drop behind them when the funnel is raised. On the bracket is rotatably mounted a conical body having round the base an enlargement, of spiral form and sharp at the commencement. The conical body is rotated by an arm, so that the underside of the spiral enlargement may pass over the bottom end of the link, securing the link. When the arm is turned in the opposite direction, the sharp end of the enlargement passes under the end of the link and lifts it clear of the projections. The arm is supported when in its raised position by a chain.

23,906. RUDDERS. R. S. BAGNALL & SONS AND A. F. FAIRBAIRN, SOUTH HYLTON FORGE, NEAR SUNDERLAND.

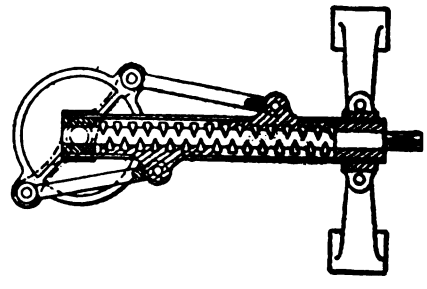
Relates to the connection of the rudder post to the rudder stock. The rudder post is connected to the rudder stock at an angle of about 45 degrees by half-lap joints, which are tied together by bolts. The meeting faces are inclined. The post is provided with an extension, to



which the rudder plate is riveted. The upper gudgeon is integral with the stock and is connected to the stern frame by a pintle, of which the head is placed sufficiently far beneath the under surface of the gudgeon to allow the stock to be raised to clear the dovetailed face between the laps, so that the rudder may be unshipped without removing the pintle.

28,259. STEERING SHIPS. H. JARMAN, LONDON.

Relates to screw steering gear for yachts, etc., of the type in which a divided nut with double screw action is employed to transmit the mo-



tion from the steering wheel to the rudderhead, The half-nuts are enclosed in a slotted supporting tube mounted at one end on the rudderhead and at the other on a fixed standard.

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